



EFFECT OF RAILWAY VEHICLE PARAMETERS VARIATION ON DERAILMENT RUNNING ON CURVED TRACK

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Declaration

I hereby declare that the work, which is being presented in this thesis entitled “**Effect of Railway Vehicle Parameters Variation on Derailment running on curved track**” is original work of my own, has not been presented for a degree of any other university and all the resource materials used for this thesis have been duly acknowledged.

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Date

This is to certify that the above declaration made by the candidate is correct to the best of my knowledge.

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**EFFECT OF RAILWAY VEHICLE PARAMETERS ON DERAILMENT RUNNING ON
CURVED TRACK**

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Aknowledgment

First, I would like to thank my God, who helps me in all my life. Secondly, I would like to express my gratitude to my advisor Mr. Tollossa Deberie for his successive look up, concerned, valuable and constructive comments during the whole period of this master thesis work. Thirdly, I thanks all the researchers whose articles referenced in the preparation of this study.

At last not least, I would like to thanks my family and my friends specially my wife Alemnesh Walelign, my brother Solomon Desta and my friend Girum Teferi for their unreserved encouragement and support they rendered to me during the entire period of this mater study.

Abstract

Railroad systems are the most widely used methods of transporting passengers in the world, so the system must be safe and comfortable. Safety is one of and critical railway system running behavior. In this paper effect of railway vehicle parameters variation on derailment running on curved track at curved critical speed was studied. The parameters are carbody parameters(mass,rolling mass moment of inertia I_{xx} , pitch mass moment of inertia I_{yy} , yaw mass moment of inertia I_{zz} and vertical mass center), secondary suspension parameters, wheelbase distance and wheel radius. Each parameters in the study had to varying 25% upper and lower of original designed value. The study has been performed by developing 41 DOFs, 4 axles and two bogie railway vehicle with SIMPACK software. First it was determined the curved track critical speed of the model vehicle on curved track scenario, when increasing speed above this maximum speed derailment occurs on the railway vehicle. After determination of the curved critical speed or maximum safe speed parameteric variation was analysed. In general terms, it was concluded that carbody mass, carbody yaw mass moment of inertia(I_{zz}), carbody height of center of mass, secondary spring longitudinal shear stiffness, wheel base and wheel radius are more sensitive to derailment, thus parameters have a roll to improve curved track critical speed and safety design of railway vehicle. Increasing or decreasing of rolling mass moment of inertia(I_{xx}) of carbody, pitch mass moment of inertia(I_{yy}) of carbody, secondary vertical damper damping and secondary lateral damper damping, decreasing secondary spring lateral shear stiffness and secondary spring vertical shear stiffness have no effect to derailment of railway vehicle running on curved track at curved critical speed.

This research analyzes only the safety running behavior of of the railway vehicle running on curved track, this means in the study was not included other running behavior such as ride confort and wear. These parametric variation may have negative effect on other running behavior. There fore decreasing yaw mass moment of inertia(I_{zz}) of carbody, vertical mass center of carbody, secondary spring longitudinal shear stiffness, bogie wheelbase, wheel radius below the original value have safe to risk of derailment and improve curved track critical speed, but to take the parameters variation as positive the effect of those vriation will investigate to other running behavior, must be improved or with in the permissible limit.

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Abbreviations

AAiT	Addis Ababa Institute of Technology
ERC	Ethiopian Railway Corporation
MBS	Multi Body System
AALTR	Addis Ababa Light Rail Transit
LRT	Light Rail Transit
DoE	Design of Experiments

Chapter One

1. Introduction

1.1 Background

Along the historical development of transportation system, railroad vehicles are the most safe, fast and widely used methods of transporting passengers. Trains have been used in commerce for more than a century, but the last three decades, in particular, have seen significant progress in rail transportation technology. Some modern trains operate at high speeds to minimize cost and transportation time. As train operating speeds increase, safety and comfort remain paramount concerns. Nevertheless, once dealing with running on curved track, important factors like ride comfort and safety are forcing to suffer from relevant degradation, which in critical cases can lead to undesired situations such as a derailment. Consequently, it is extremely important for the railway companies to consider the ride comfort and safety issues during in designing, manufacturing, testing and operation process. Therefore, modern trains are complex mechanical systems that can be better analyzed and designed using modern computational mechanics techniques.

The multi-body simulation software is a key method for design; homologation and research developing detailed computer models of railroad vehicles is necessary to study vehicle performance, to improve existing designs and develop new ones, and to develop safety guidelines for different operating and loading conditions. Computer dynamic simulations have, in fact, come to play an integral role in railroad vehicle performance, in safety and accident evaluation, and in the design of new vehicles [2].

Many bodies comprise the train system and so it has many degrees of freedom [1,3]. The bodies that make up the vehicle can be connected in various ways and a moving interface links the vehicle to the track. This interface involves the complex geometry of the wheel tread and the rail head and non-conservative frictional forces generated by relative motion in the contact area [3]. Numerical solution is necessary, lots of multi-body-system (MBS) software supply possibility for quick modeling and simulating of railway vehicle dynamic performance.

The MBS simulation software is not only the most feasible method for the prediction of the safety/derailment, ride comfort, wears, fatigue and noise behaviour of rail vehicle and truck, but also many other common and advanced tasks can today handled [5]. For general vehicle systems, MBS software simulations and analysis represent an efficient way to accomplish the growing need for cost and time reduced development cycles of new vehicles as well as to optimize/improve the railway parameters in order to make it more safe and comfortable for the passengers [3,4,5].

[6] Railway lines construction in Ethiopia was first starting in October 1897 from Djibouti in the period of Emperor Menelek II. The first commercial service began in July 1901, from Djibout to Dire Dawa. By 1915, the line reached Akaki, 23 kilometers from the capital, and two years later came all the way to Addis Ababa. The railway links Addis Ababa, the capital of landlocked Ethiopia, to the port of Djibouti in coastal Djibouti. Maintenance shops along the line are located in Dire Dawa, which grew up as the Imperial Ethiopian Railway depot for nearby Harar. The single-track 781 km railway has a 1,000 mm gauge, most of it on Ethiopian territory, and about 100 km in Djibouti. Today the old line is out of service, but a new standard gauge railway system above 5000Km to different directions of the country is under construction for freight and passenger transportation. In addition, in order to solve Addis Ababa city Public transport diverse problem the city administration is under construction of 34.25 km (North-South line 16.9 km and East-West line 17.35 km) Light Rail Transit (LRT) system. Now a day in Ethiopia all railway system is under construction, but in this thesis simple and reliable Railway vehicle model developed and analysed the effect of Railway Vehicle Parameter Variation on Derailment of the Railway Vehicles Running on Curved Track at Curved Critical Speed obtained through Multi-body simulation.

In modern railway industry the simulation of the behaviour of railway vehicles has become an important validation method, it is much cheaper, easier and quicker than doing it with a real model. Modern simulation packages such as VAMPIRE, NUCARS, GENYSIS and SIMPACK offer modelling and simulation elements that are highly adapted for standard and unusual simulation scenarios [9]. To compute effect of Railway Vehicle Parameter variation on Derailment of Railway Vehicles Running on Curved Track at Curved Critical Speed of LRT system need the development of model with those MBS software, simulating and analysing the effect of LRT system, it is possible to use the great amount of the above available software. SIMPACK will be used to perform this Master Thesis.

For designers it is important to have better understanding on effect of railway vehicle parameters variation on derailment of railway vehicles running on curved track. This study give an information about which railway vehicle parameters variations more sensitive to the risk of derailment, which parameters improve critical speed and which parameter variation have no effect on risk of derailment running on curved track. Decreasing risk of derailment in addition to safety help to decrease wheel rail contact wear, which minimize maintenance cost.

Furthermore, give direction in the future to evaluate the performance when new vehicle is designed, to optimize the existing railway vehicle system which vehicle component needs optimization work in terms of safety and to prepare railway system speed profile as well as to fix the maximum allowable speed on curved track by SIMPACK MBS software in Ethiopian Railway Corporation (ERC).

1.2 Purpose

The purpose of this thesis is to analyse the effect of railway vehicle parameters variation on derailment running on curved track. The analysis done by creating simple but reliable railway vehicle in the SIMPACK MBS software with curve operational track scenarios.

1.3 Statement of Problem

Railway is most widely used mass passenger transportation system in the world, so the system must be safe and comfortable to passengers. Two unique features of operation, running the train metal wheels on metal track and guidance by the track lead to the railway system is one of the most complex dynamical systems in engineering. These dynamic interactions between the wheel and the rail in railway systems result very complex running behavior. The major problem in designing, manufacturing, construction and operation of the rail system is due to the complex running behavior of railway system improper design and performance evaluation result to railway derailment, wear and poor ride comfort. Therefore, for good new railway vehicles design and to perform improvement work on exiting railway vehicle for objective function of running behavior performance need to have a better knowledge on effect of railway vehicle parameters variation on safety and ride comfort.

This study give an information about which railway vehicle parameters variations more sensitive to the risk of derailment, which parameters improve critical speed and which parameter variation have no effect on risk of derailment running on curved track. Decreasing risk of derailment in addition to safety help to decrease wheel rail contact wear, which minimize maintenance cost.

Hence, designing and manufacturing of safe new railway vehicle system in consideration of the safety function and improving of existing railway vehicle system curved critical speed are major problem areas that caused this research work to be carried out.

1.4 Objective of the Study

1.4.1 General Objectives

The general objective of the thesis, as the title already states are, to develop a simple but reliable conventional vehicle Model in multi-body simulation (MBS) software SIMPACK , simulate and analyze effect of Railway Vehicle Parameters Variation on Derailment of Passenger Railway Vehicles Running on Curved Track at Curved Critical Speed.

The parameters are carbody parameters(mass, rolling mass moment of inertia I_{xx} , pitch mass moment of inertia I_{yy} , yaw mass moment of inertia I_{zz} and vertical mass center), secondary suspension parameters, wheelbase and wheel radius.

1.4.2 Specific Objectives

Specifically, this paper has the following objectives:

- Study multi-body system dynamics software, SIMPACK and presented a procedure for developing railway vehicle.
- Perform preload Simulation to determine a static equilibrium position.
- Determine critical speed of the railway vehicle on given curved track scenario.
- Determine the ratio of lateral and vertical force (Y/Q) at wheelset as indicator of wheel climbing derailment.
- Determination of Sum lateral force of rail (sum Y) indicator of track shifting force lead to track irregularity and in the latest case to a derailment.
- Conduct Parametric analysis by varying rail parameter and study the effect of each parameter on derailment.

1.5 Scope and Limitation of the Study

1.5.1 Scope of the Study

This thesis covers only developing simple but reliable vehicle model and simulates the model, analyses the effect of railway vehicle Parameters Variation on Derailment of Railway Vehicles Running on Curved Track at Curved Critical Speed.

1.5.2 Limitation of the Study

A computer model of a railway vehicle can be constructed and run on typical or measured track in a virtual environment, and a wide range of possible designs or parameter changes can be investigated. Outputs from the model can be set up to provide accurate predictions of the dynamic behavior of the vehicle and its interaction with the track. In this thesis due to lack of sufficient materials and data the limitations present as follow:

- Cross wind force are not taken in to account but it does influence on rail vehicle derailment.

1.6 Organization of the Paper

The paper organized by five chapter, in chapter one it discusses the background of railway system, discusses statement of problem and specifies purpose of the research, describe general and specific objectives, scope and limitation of the study. In chapter two discuss theory of derailment and multi-body system modelling and simulation software and presents related literature review relevant to this work. The third chapter describes modeling materials, conditions, present the methodology of modeling railway vehicle by SIMPACK MBS software with vehicle data and define each modeling parameters, after modeling perform preload and static equilibrium calculation, which used to perform each force elements in equilibrium and to determine the equilibrium position of the full model in the three directions . At the end set solver setting for time integration and describe SIMPACK post processing and present final result expectations. The fourth chapter deal with the interpretation and explanation of the findings of the study, that is the developed model in chapter three SIMPACK experimetal simulation performed and the results discuss to the objective function. In fifth chapter present a summary of findings. Besides conclusion and

recommendations, the implication of findings should be explained here and additional research areas should be suggested as well as future projection based on the study.

Chapter Two

2. Literature Review

2.1 Theory of Railway Derailments

The railway vehicle running along a track is one of the most complex dynamical systems in engineering. One of the most important behaviors of a railway vehicle at high speeds (especially on curved tracks) is safety/derailment. Without any kind of hesitation, this parameter has to be scrupulously studied until acceptable levels are achieved.

Derailment has always been one of the major concerns for railway operations since the first day of wheels running on rails. Railway technologies have advanced significantly in recent years and safety levels are highly improved compared with the early days and compared with other transport modes, but derailments unfortunately still occur.

Railway derailments due to loss of the lateral guidance at the wheel and rail interface may be classified into four major causes: wheel flange climb, gauge widening, rail rollover, and track panel shift, based on the ways that wheel – rail lateral constraints are lost [18].

- A. Wheels climbing onto the top of the railhead then further running over the rail cause wheel flange climb derailments. Wheel climb derailments generally occur in situations where the wheel experiences a high lateral force combined with circumstances where the vertical force is reduced on the flanging wheel. The high lateral force is usually induced by a large wheelset angle-of-attack. The vertical force on the flanging wheel can be reduced significantly on bogies having poor vertical wheel load equalization. Flange climb derailments generally occur on curves.

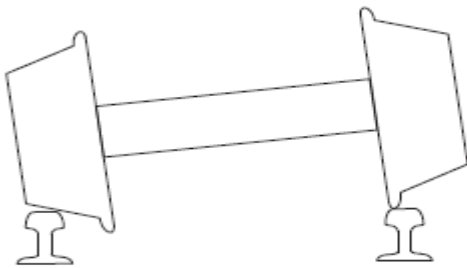


Figure: 2.1: Wheel Climb

- B. Derailments caused by gauge widening usually involve a combination of wide gauges and large lateral rail deflections (rail roll), as shown in Figure: 2.2: Large lateral forces from the wheels act to spread the rails in curves.

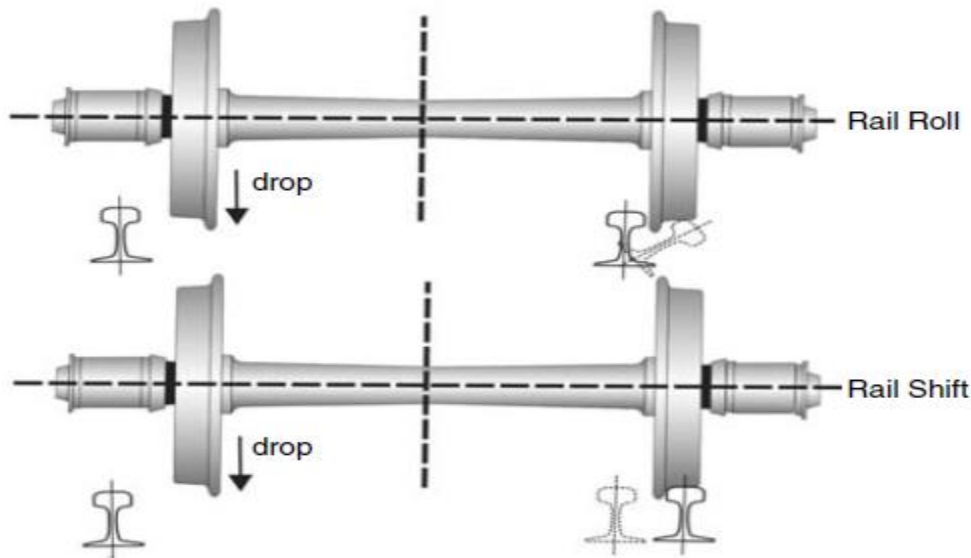


Figure:

Figure: 2.2: Gauge widening derailment

- C. Both rails may experience significant lateral translation and/or railhead roll, which often causes the wheel to drop between rails. Figure:2.2: was produced based on a photo of an actual derailment caused by rail rollover, in which the wheel fell between rails and the outer rail was rolled over.

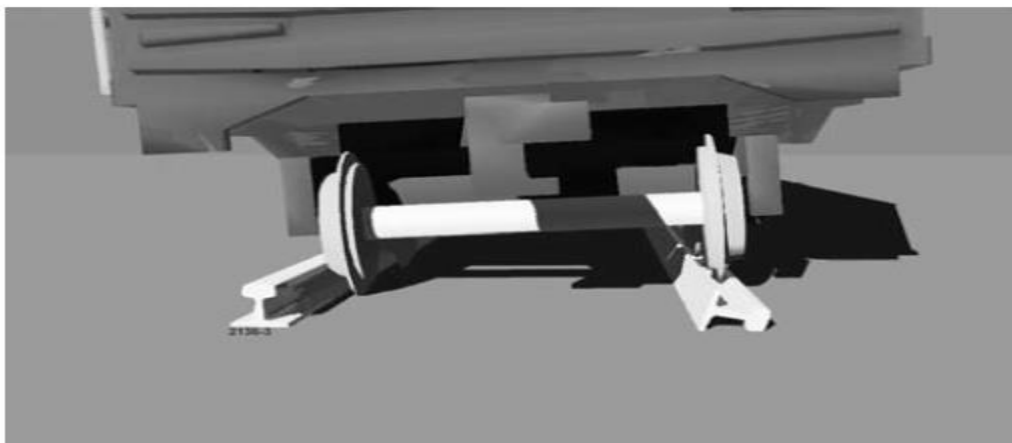


Figure:2.3: wheel falls between the rails, while the outer rail is rolled over.

D. Track panel shift is the cumulative lateral displacement of the track panel, including rails, tie plates and ties, over the ballast, as shown in Figure:2.4:. A small shift of these components may not immediately cause the loss of guidance to bogies. However, as the situation gradually depreciates to a certain level, wheels could lose guidance and drop to the ground at some speed. The derailments caused by track panel shift usually result in one wheel falling between the rails and the other falling outside of the track.

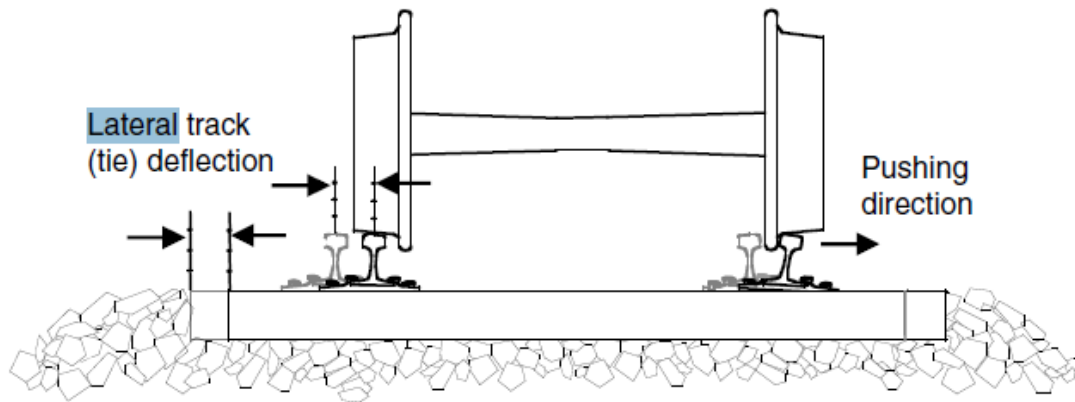


Figure: 2.4: Lateral track panel shift

2.2 Safety assessment of Rail Vehicle

According to the EN-14363 standard [CEN (2005)], the safety of a rail vehicle is assessed by means of two quantities: the track shift forces and derailment coefficient.

2.2.1 Track shift forces

The first parameter that quantifies safety deals with the lateral forces created due to the wheel-rail contact as the vehicle runs over the track. This is particularly important because a high value of track shift forces leads to track irregularities (which increases the maintenance costs) and in the latest case to a derailment. This derailment parameter assessed the derailments phenomena occur due to gauge widening or railhead roll or track pannel shift.

Equation (2.1) defines how to calculate this value for the leading wheelset [CEN (2005)]:

$$\sum 2Y \leq K_1 \left(10 + \frac{2Q}{3} \right) \quad (2.1)$$

With k_1 taking account of greater variations in geometrical dimension and of the state of vehicle maintenance ($k_1=0.85$ for freight wagon and $k_1=1$ for passenger vehicles)[18] and $2Q_0$ is the mean axle load of the vehicle defined by

Equation 2.2:

$$2Q_0 = \frac{m_{veh}g}{n} \quad (2.2)$$

Where m is the mass of the vehicle, g is the gravitational force and n is the number of axles of the vehicle.

In this study the track shift force in SIMPACK post process present as sum lateral rail force and the value of this output parameter analyse whether increase or decrease with railway vehicle for each parameter variation.

2.2.2 Derailment coefficient

Another factor that must be taken into account while analyzing the railway safety is the parameter that quantifies the risk of derailment. It is called derailment coefficient and is defined with equation (2.3) [CEN (2005)]:

$$\left(\frac{Y}{Q} \right) \quad (2.3)$$

Where, Y and Q represent the lateral and vertical forces for the wheel–rail contact respectively and the maximum limit value for Y/Q is 0.8 [21].

In this study the derailment coefficient in SIMPACK post process present as similar to the above equation 2.3 (Y/Q) and the value of this output parameter analyse whether increase or decrease with railway vehicle for each parameter variation.

2.3 Multi Body System Software

In modern railway industry the simulation of the running behaviour of railway vehicles has become an important method for design, homologation and research in the field of railway and similar vehicle. Multi Body system simulation is easier and quicker than doing it with a real model and help to avoid many costly measurements and tests as well as unexpected occurrences when the vehicle is in operation.

Commercial computer tools for railway vehicle have been available already from 1980s. Now a day there are a large numbers of 'handmade' in-house tools in the industries and at universities, as the same time the number of users and the mathematical and numerical complexity of the software has dramatically increased [5]. From the number of developed simulation tools there may be available a thousand of licensed commercial high-end MBS tools worldwide.

Without MBS tools, as most of companies in previous, engineers should spend most of their time in establishing differential equations and solving them by themselves. They had spent less time on solving faced practical problems. With MBS tools as now, almost all researches/engineers can focus on solving the problem by changing various conditions and understanding what going on physically.

A MBS model is a mechanical model constructed from separate bodies which have their own mass and inertia properties. The separate bodies are connected each other by joints, kinematic constraints and force elements. The set of the joints and constraints defines the number of the system degrees of freedom(DOF). The movements within the available DOFs affected by the force and torque elements. The system is completed by sensors and excitation elements for the designed outputs.

From various modern simulation packages ADAMS/VI-Rail, MEDYNA, A'GEM, MODKAT, ROBSTRAN, SAMS, Universal Mechanisms, VOCO, VOCODYM, VAMPIRE, NUCARS, GENYSIS and SIMPACK offer modelling and simulation elements that are suitable for railway simulation and highly adapted for standard and unusual simulation scenarios [9,5].

Modern Multi Body System software applicable or useful in railway sector for vehicle and component manufacturers, railway operators, railway homologation authorities, consultants and engineering service providers and universities and related research institutes.

The MBS simulation software is the one which provides the ability to represent system details, all important interaction while allowing for fast computations during long simulation times. This computational approach applied in railway dynamics is described in three important modules: the vehicle; the track and the interaction between. The coupling of these formulations, which describe the vehicle, track and their interaction, is governed by differential-algebraic equations (DAE). For the dynamic analysis, this system of equation is systematically solved and then integrated on time. This constitutes a pure mathematical problem which can be solved with numeric methods that

involve sparse matrix solvers to solve the system of equation, and the Gear multistep method for the integration of the differential equations [20]. Most of MBS simulation package generate these equation automatically from interactively created model of the system.

In this project, one of the most well-known software accepted by industrial and academic communities called Multi Body Simulation (MBS) software SIMPACK rail module is used. The validation of SIMPACK simulation result with measured test result obtained from [15]. The comparisons between simulation and measurement were carried out for all vehicle models and model configurations under the same conditions over selected track sections of the test runs, called validation exercises. One validation exercise consists either of a part of straight track or of one curve passing scenario, including both transitions and parts of straight track. The selected 17 validation exercises represented all 4 track zones according to EN 14363:2005 : straight track and very large curves were represented by 5 sections, large radius curves ($R > 600$ m) by 2 sections; 4 sections were from small radius curves ($400\text{m} \leq R \leq 600$ m) and 6 from very small radius curves ($250\text{m} \leq R < 400$ m). They were from three countries: Germany, Italy and Switzerland.

The article presents the investigations conducted by the whole project team and the proposed approach for the model validation. The validation results are illustrated with examples of two vehicle models, both developed in the MBS simulation tool Simpack: DB passenger coach Bim prepared by Bombardier Transportation. This vehicle is a single-deck passenger coach with two 2-axle bogies of Minden-Deutz design and Freight flatbed wagon Laas modelled by Alstom. This wagon unit consists of two 2-axle wagons with UIC double link suspension, connected by a short coupler to one vehicle unit.

The models of locomotive DB BR 120 and DB passenger coach Bim were successfully validated. Neither a model configuration of the freight vehicles nor a model configuration without the usage of measured track irregularities could be validated. This result demonstrates the difficulties of modeling freight vehicles with large uncertainties of friction suspension parameters and stresses.

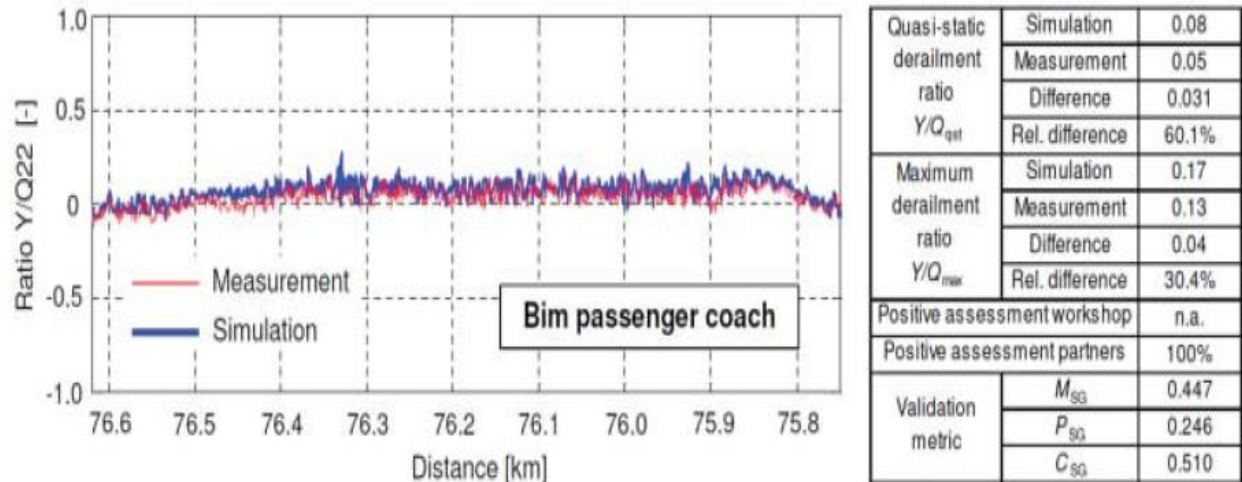


Figure: 2.5: Germany, curve radius 580 m, speed 110 km/h: positive assessment by partners in contrast to high error factors.

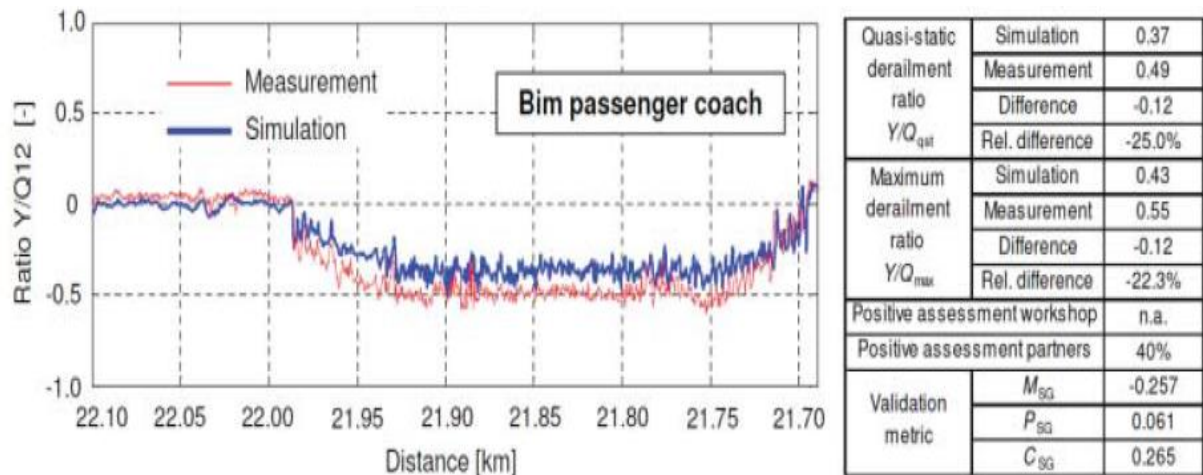


Figure: 2.6: Validation examples Exercise F1.12, Italy, curve radius 295 m, speed 76 km/h.

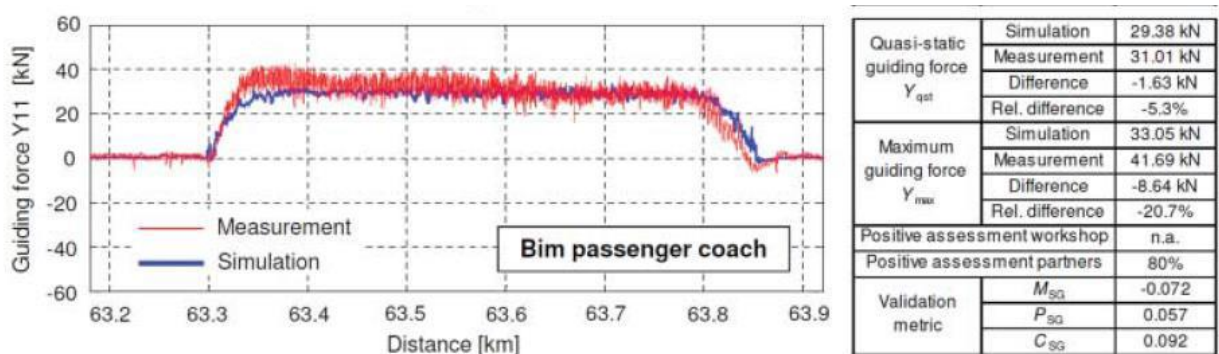


Figure: 2.7: Validation examples Exercise F1.01, Germany, curve radius 282 m, speed 68 km/h.

2.4 Related Research Literature Review

As a starting point to understand the behavior of a rail vehicle and the corresponding effects on the ride comfort, safety and wear, it is necessary to investigate different linear and nonlinear system dynamic responses. The critical hunting speed as the origin of the instabilities of a rail vehicle as well as the effects of the wheel conicity, the wheel-rail contact and the track imperfections have been studied thoroughly in [Fan and Wu (2006)]. To overcome the negative effects of such parameters on the above mentioned objectives functions, several suspension systems and control strategies have been proposed.

[4] Studied. 50 DOFs one-car railway vehicle model with two bogies through Multi-Body Simulation software SIMPACK to Improving/ optimize the dynamics behavior in terms of ride comfort ride, safety and wheel-rail wear objective functions. The optimization problems are focused on the optimization of the bogie primary suspension springs and dampers components with respect to the wheel-rail wear objective function while ride comfort and safety are taken as thresholds or within the admissible limits.

[3] Designing railway model with ADAMS MBS software and afterward, performing some Simulations. Preload Simulation will be done as well as Linear Simulation, Stability Simulation and Dynamic Simulation. In the end, an analysis of the results will be obtained.

[10] Manchester Benchmark investigated the dynamical performance analysis for TGV 001 locomotive vehicle using tool (VIA) and the results obtained are compared with SIMPACK simulation Package for the same vehicle model, at the same operating condition.

[9] Investigated the effect of vehicle dynamic behavior Rolling Contact Fatigue of Swedish iron ore freight wagon using MBS software GENSYS. Validate the model by comparing experimental data and simulation results.

[14] By developing comprehensive MBS simulation model of a four-axle high-speed passenger rail vehicle investigations and validated pre- and post-derailment with authentic derailment incidents. The rail vehicle analysis tool GENSYS [2] is used in all MBS time-domain simulations.

[16] Assess the influence of the inertial properties of railway vehicles on their dynamic behavior, Safety, ride quality and track fatigue. The studies analyse a degree of sensitivity of the masses, moments of inertia and heights of the center of gravity of the carbody, the bogie and the wheelset on dynamic behavior of railway vehicle by SIMPACK MBS software. In general terms, it was concluded that the most sensitive inertial properties are the mass and the vertical moment of inertia, and the least sensitive ones the longitudinal and lateral moments of inertia.

[17] Loaded freight vehicle with two three-piece bogies is modeled using the accessible mathematical software MATLAB and the results are compared with its corresponding ADAMS/Rail dynamic MBS simulation model, where similar derailment factors are encountered for the freight vehicle. Both models reveal that the possibility of derailment increases immediately after entering and at the end of the curve – signifying the beginning and the end of the curve as two major points for derailment.

[18] Presents critical speed derivations and kinematic descriptions of motions and instabilities on curved and straight track using fundamental physical principles. The study showed that the best practice for increasing speed for all traffic was to make the curve gentler (increase the radius).

2.5 Research Gap and Conclusion

The above discussed researches have a research gap to addressing effects of all railway vehicle parameters such as secondary suspension parameters, wheelbase and wheel radius variations on risk of derailment on curved track. Therefore, this paper analyse the effect of all parametric variations on derailment running on curved track by using efficient and validated[15] SIMPACK MBS simulation software and the result will give better input for understanding, design, optimization and homologation process of rail vehicles interms of safety function.

Chapter Three

3. Materials, Conditions and Methods

3.1 Materials

3.1.1 Railway Vehicle and Track

The vehicle model used in this study represents a conventional passenger rail car with two bogies, with the carbody resting on the elastic elements of the secondary suspension shown in Figure: 3.1. It follows the track centerline with the speed of the vehicle. The track is rigid and have both straight and curved standard gauge track. Design data of the vehicle are presented in appendix: A and design characteristics of the track are presented in appendix: A

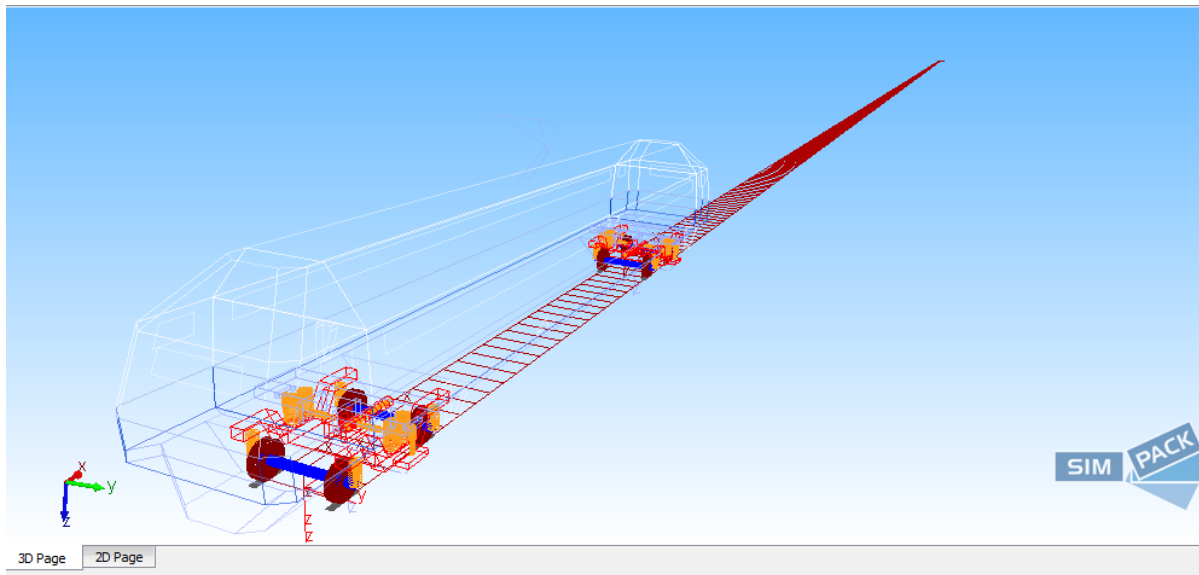


Figure: 3.1: Rail Vehicle Model

3.1.2 Simulation Software

To compute effect of Railway vehicle parameter variation on safety(derailment), first develop a simple but reliable railway passenger vehicle model in multi-body simulation (MBS) software SIMPACK modules version SIMPACK 9.6 (32bit). SIMPACK Rail provides all the necessary functionality for creating and working with highly detailed and accurate railway dynamics models.

SIMPACK Rail allows:

- Improve the design before building the physical prototype.

- Analyze design changes much faster and at lower cost using physical prototypes.
- Realize a variety of analysis, so much easier than changing the instrumentation required in a real model.

3.2 Conditions

The simulation performed on a straight and curved standard gauge rigid track with speed start from 1m/s increase to 25m/s running on the straight track and decrease the speed at the entrance of curved track to curved critical speed 20m/s. Speed increase above the curved critical speed on the curved track derailment occur on the rail vehicle. In addition the following assumption has taken

- Axle box was assumed as a wheel set's integral part, hence its weight was combined with the wheel set's mass
- All wheel profile and axle were consider identical
- Friction did not exist between axle and bearing
- Bogie and car body component masses were considered to be rigid
- Ideal track is taken which means excitation force due to irregularity of the track not considered
- Effect of active suspension like train tilting mechanism will not be include

3.3 Methods

The MBS simulation package is the one which provides the ability to represent system details, all important interaction while allowing short simulation times. This computational approach applied in railway dynamics is described in three important modules: the vehicle; the track and the interaction between Railway vehicles and roads represent the most important types of the application of the MBS approach. A MBS can be defined as a collection of rigid and/or flexible bodies interconnected by kinematic joints and/or force elements [20]. The relative motion between the bodies forming the MBS is controlled by the kinematic joints. On the other hand, the force elements represent the internal forces, which are produced due to the relative motion between the connected bodies.

General procedure of modeling and simulating of railroad vehicle-track system [9] described in figure: 3.2

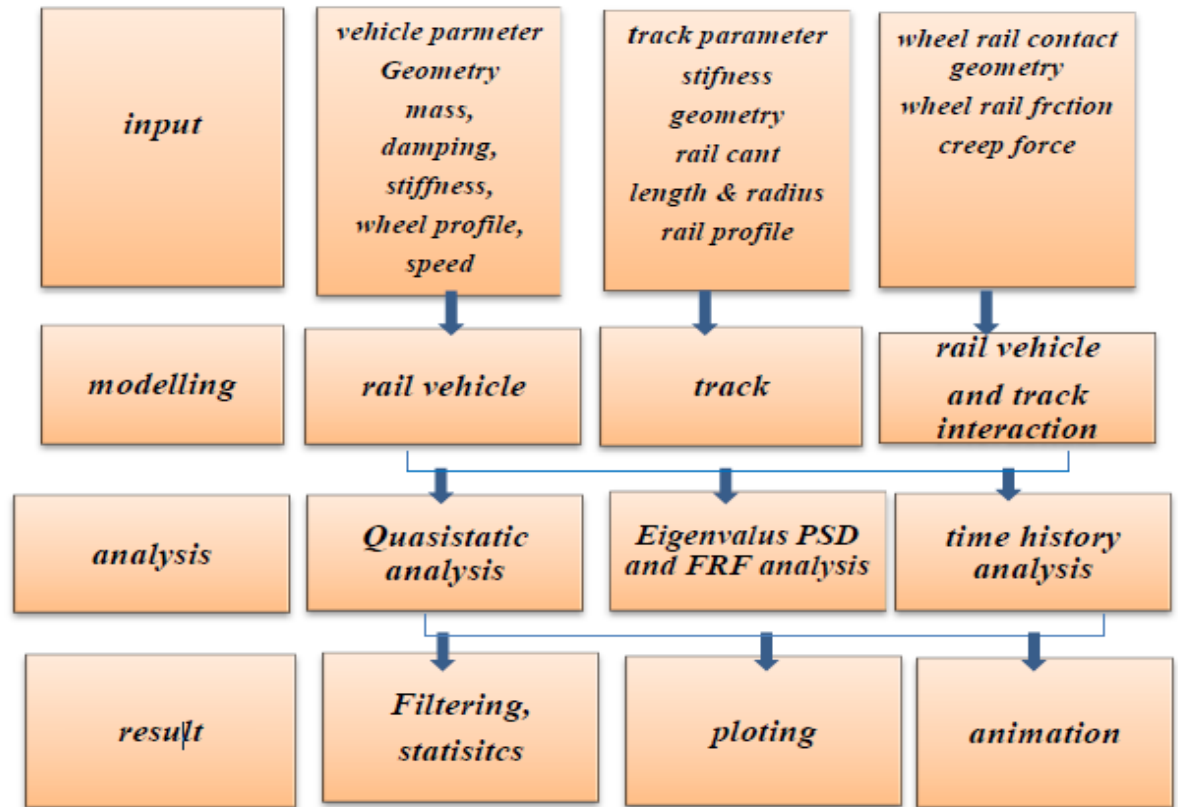


Figure: 3.2: General MBS modelling procedure

3.3.1 Model Developing Procedure

According to various rail vehicle norms, a reference system, xyz, was defined, the z-axis points downwards and the y-axis points to the right when looking in forward direction in order to create the appropriate SIMPACK model wheel/rail model. The rail-wheel contact and many related algorithms, plots etc. relay on this orientation [8].

The vehicle main components (carbody, bogie frames and wheelsets) were modeled as rigid bodies, connected to each other by means of non-linear springs and dampers that characterize the primary and secondary suspensions. The procedure of developing rail vehicle model by using rail-vehicle data is start from creating wheelset and track to develop the bogie and then developing the complete vehicle. Figure:3.3: is the kinematic model of the complete developed model of rail vehicle which shows the kinematic interaction of the car body with the inertia fixed reference frame, rear and front bogies \$S_Rear and \$S_Front, respectively. In this rail vehicle development

procedure naming of components and parameters are specific, but any one who use SIMPACK software to develop model use the naming as it is or different naming.

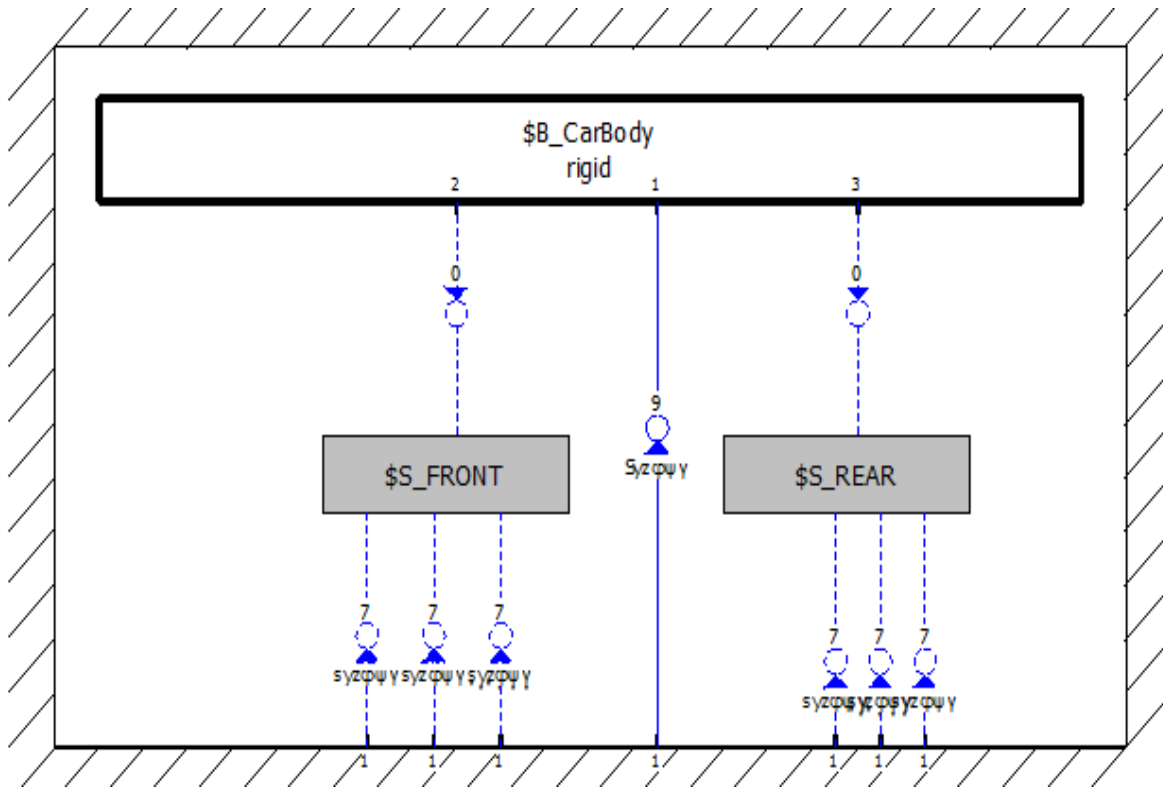


Figure: 3.3: Topological diagram of the vehicle model, Degrees of freedom: x, along the track; y, lateral; z, vertical; α , roll; β , pitch; γ , yaw.

3.3.1.1 Development of Bogie

First step is create a new SIMPACK model using the 'General' template and switch the gravity direction by changing the sign in z direction from minus to plus. Rotate the view so that the x-axis points forward, the y-axis to the right and the z-axis downwards. Then save the created model. Each input data was parameterized with the aim of easing the variation of its features during the derailment analysis. For each input parameter assign SubVar shown in Figure: Appendix: A.1:

Front Wheelset Pair

Front Wheelset Axle shaft: The first step is creating front wheelset axle shaft. Rename the created body '\$B_WS_F' or with other any name start with \$B_. Fill the body properties dialog

box figure:3.4 with the given wheelset mass and inertia properties data by the assigned SubVar . Mass and inertia properties of the wheelset assumed including the axle boxes: $\$_{Wheelset_Mass}$ =1813 kg, Inertia $\$_{Wheelset_Ixx} = I_{zz} = 1120 \text{ kgm}^2$, $\$_{Wheelset_Iyy} = 112 \text{ kgm}^2$ for a wheelset. Primitive name to ' $\$P_WS_Axle$ ', Type 'Cylinder', $\$_{Axle_Length} = 2.0\text{m}$, $\$_{Axle_Diameter} = 0.16\text{m}$, Nnumber of planes = 50.

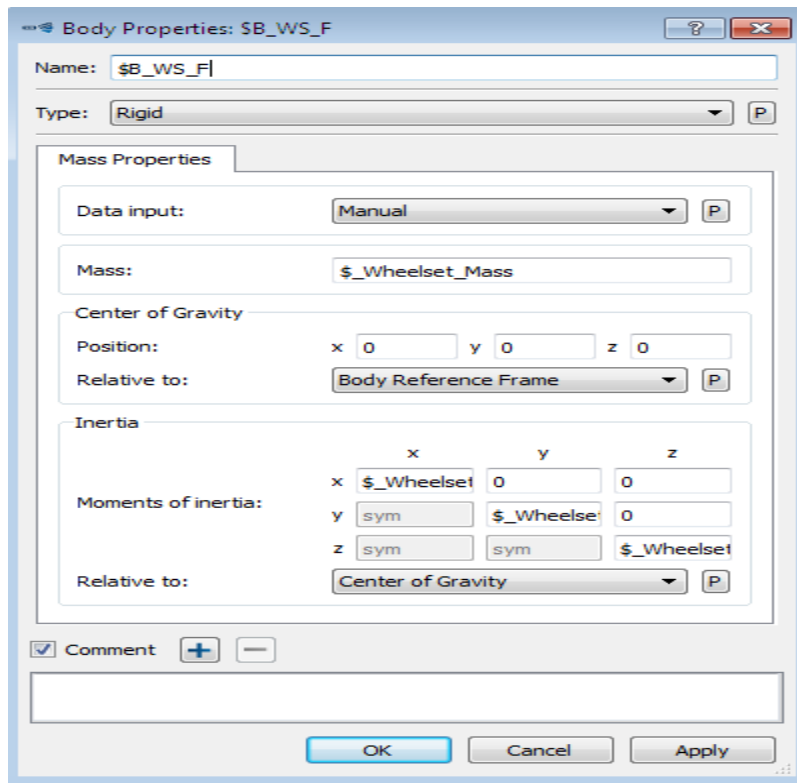


Figure 3.4 Wheelset Body Property Box

Add a standard straight Track by Create Track element - $\$Trk_Track$, Type – Rail, Reference baselength in the Superelevation group box to 1.5 m. (i.e. the approximate distance of the rail heads) and Straight track length STR- 1670 m

Front Wheelset Joint: Their movement is only restricted by the contact to the rails. It has six degree of freedom. Joint type of the wheelset - General Track Joint, type 7 and set the z position state to ' $\$_{Axle_Z} = -0.33\text{m}$ '. This will lift the wheelset shaft up to its nominal height above the track. Fill initial longitudinal position of front wheelset with $Wheelbase/2$ i.e $\$_{Wheel_Base}/2 = 1.115\text{m}$ From reference makere(Isy).

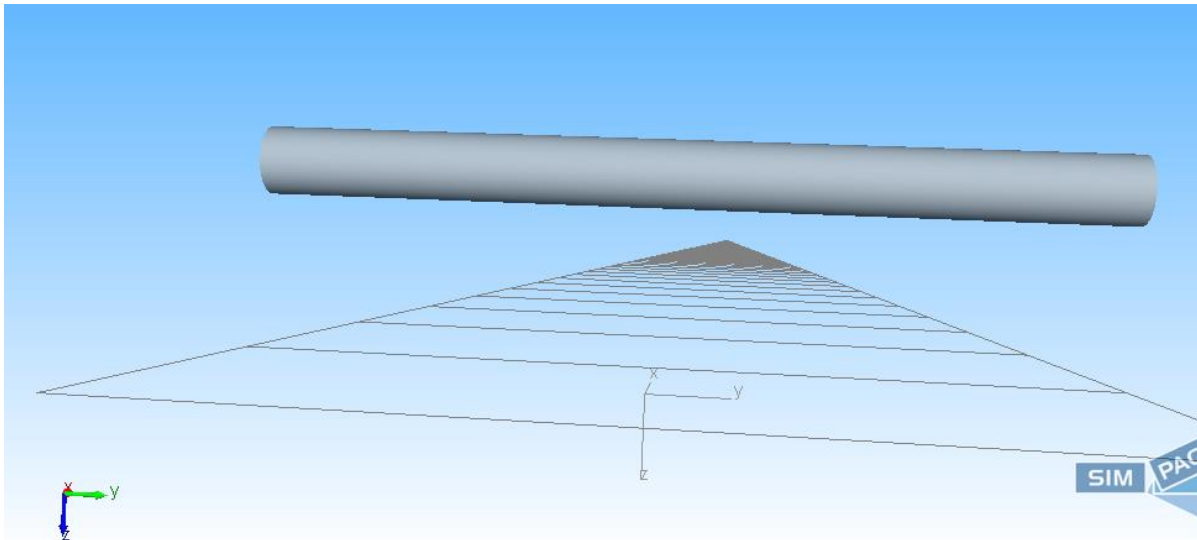


Figure 3.5 Axle Shaft & Track Layout Model

Add Front Rail-Wheel Pairs: The Rail-Wheel Pairs are the central data container and manage all modeling elements related to the rail-wheel contact. One for each rail-to-wheel pairing, i.e. two for a wheelset is required.

Create Right-hand Rail-Wheel Pair name - '\$RWP_FR' and fill the Rail-Wheel properties dialog box as follow:

Wheel : Set Wheel Carrier Body and Wheel Body with \$B_sWS_F, set Wheel Marker with \$M_WS_F, set Wheel rotatory JointJoint with \$J_WS_F, set state of joint with Gam: Pitch angle : this is the state that represents the rotational velocity. Set Wheel Radius with \$_Wheel_Radius=0.33 and fill Lateral wheel distance, this is the lateral distance from the wheelset's center (the wheel Marker) to the wheel profile reference, or taper line which is 0.75 m.

Rail : Set the Lateral rail distance method to 'explicitly set' and adjust track gauge i.e Lateral rail distance method - 0.753; Rail cant- 1:40, Tangential Forces Here, it needs a Rail-Wheel Contact element that provides the tangential (creep) forces. The FASTSIM algorithm is used with an initial discretization of 11×11 elements, Friction coefficient - \$_Friction=0.4, the parameterization of the rail-wheel contact is now finished – now click Apply to activate the element creation buttons. Finally, click Create Elements and then OK. Right side Rail-Wheel pair created

Create Left-hand Rail-Wheel Pair name - '\$RWP_FL': Now copy and paste the '\$RWP_FR' Rail-Wheel Pair in the Model Tree. Rename the new element to '\$RWP_FL', then open the Rail-Wheel Pair dialog, switch the Side w.r.t. Track forward direction to Left and click again Apply, Create Elements and OK. Not all geometry data of the new Rail-Wheel Pair are mirrored to the left side and the left rail profile from the Rail element is used. The Create Elements button has created all necessary Modeling Elements required for the rail-wheel contact simulation and their visualization and the result output.

Finaly Create Front Wheelsets: Wheelsets relate one left-hand and one right-hand Rail-Wheel Pair to each other and contain the data that belongs to this pair. Create a Wheelset and name it '\$RWT_F'. Enter the Right-hand Rail-Wheel Pair and the Left-hand Rail-Wheel Pair. SIMPACK will internally check the data and create some plots. User defined Contact Quasilinearization coefficient (Equivalent Conicity $\lambda=0.186$, Roll angle coefficient $\sigma=0.0372$ and Contact angle coefficient $\epsilon=15;81$). Again, click Apply to activate the element creation buttons. Finally, click Create Elements and then OK. This will create a View and a Result Element related to this Wheelset. Lastly the created wheelset looks like Figure :3.6

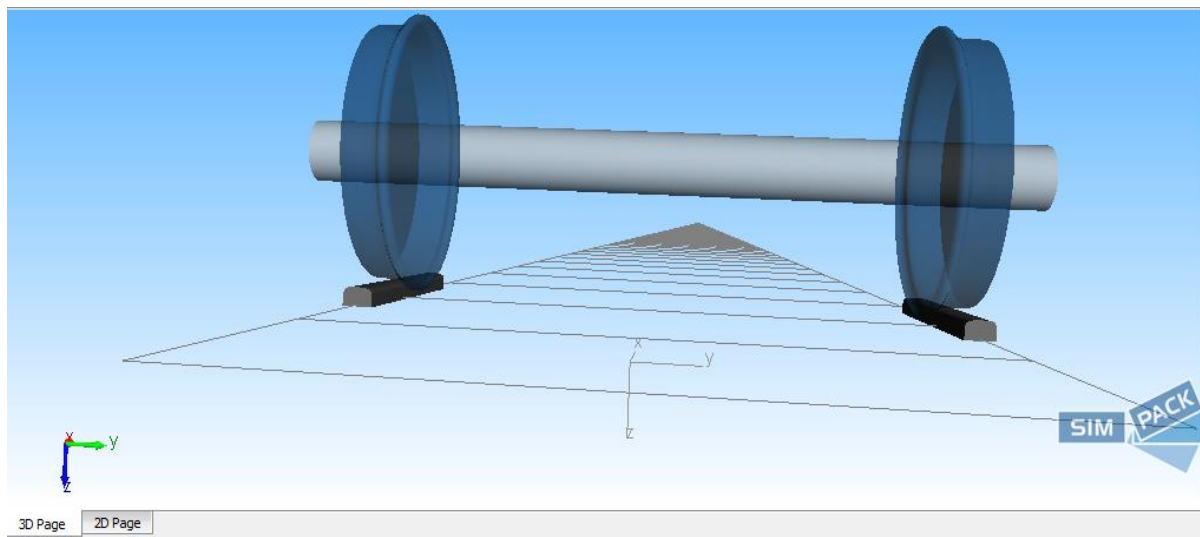


Figure 3.6 Complete Wheel & Rail Model

Rear Wheelset Pair

To develop the Rear Wheelset Pair create a second body and rename the body as \$B_WS_R. And follow all steps as Front Wheelset pair developed except setting Rear Wheelset Joint.

The Rear Wheelset pair is beyond the Front Wheelset pair with Wheelbase distance. Therefore Initial Longitudinal position of Rear wheelset is negative of Wheelbase/2 i.e $-\$_{Wheel_Base}/2 = -1.115$ From reference marker (Isy).

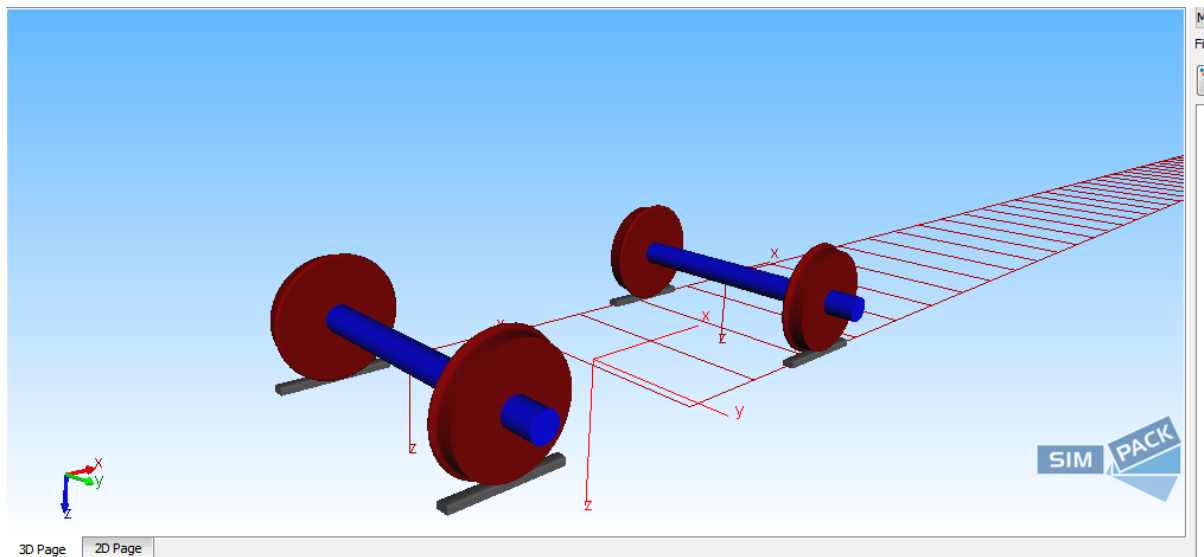


Figure:3.7: Front and Rear Wheelsets Pairs

Bogie Frame

To create the Bogie frame Create a third body and rename as \$B_frame. Fill Mass and inertia properties of the Bogie Frame assumed including all material and accessories on the Bogie frame. $\$_{Frame_Mass} = 2615$ kg, Inertia $\$_{Frame_Ixx} = 1722$ $\$_{Frame_Izz} = 3076$ kgm², $\$_{Frame_Iyy} = 1476$ kgm²

Geometry Primitive name to 'SP_Frame_side', Type '22: Wheel Rail Bogie', Length L1=3.2, Length L2=1.2, Height H1=-0.3, Height H2=0.2, Width B1=0.25 and Distance Right Left=2.0m. And Z- position=-0.28

All other Bogie Frame Geometry Primitives are developed with Cylinder, Cuboid and Sphere type with appropriate dimension and position.

Bogie Frame Joint: The Force Element between Bogie frame and Wheelsets only restricts their movement. It has six degree of freedom. Joint type of the Bogie - General Track Joint, type 7 and leave all parameter as it is or set zero.

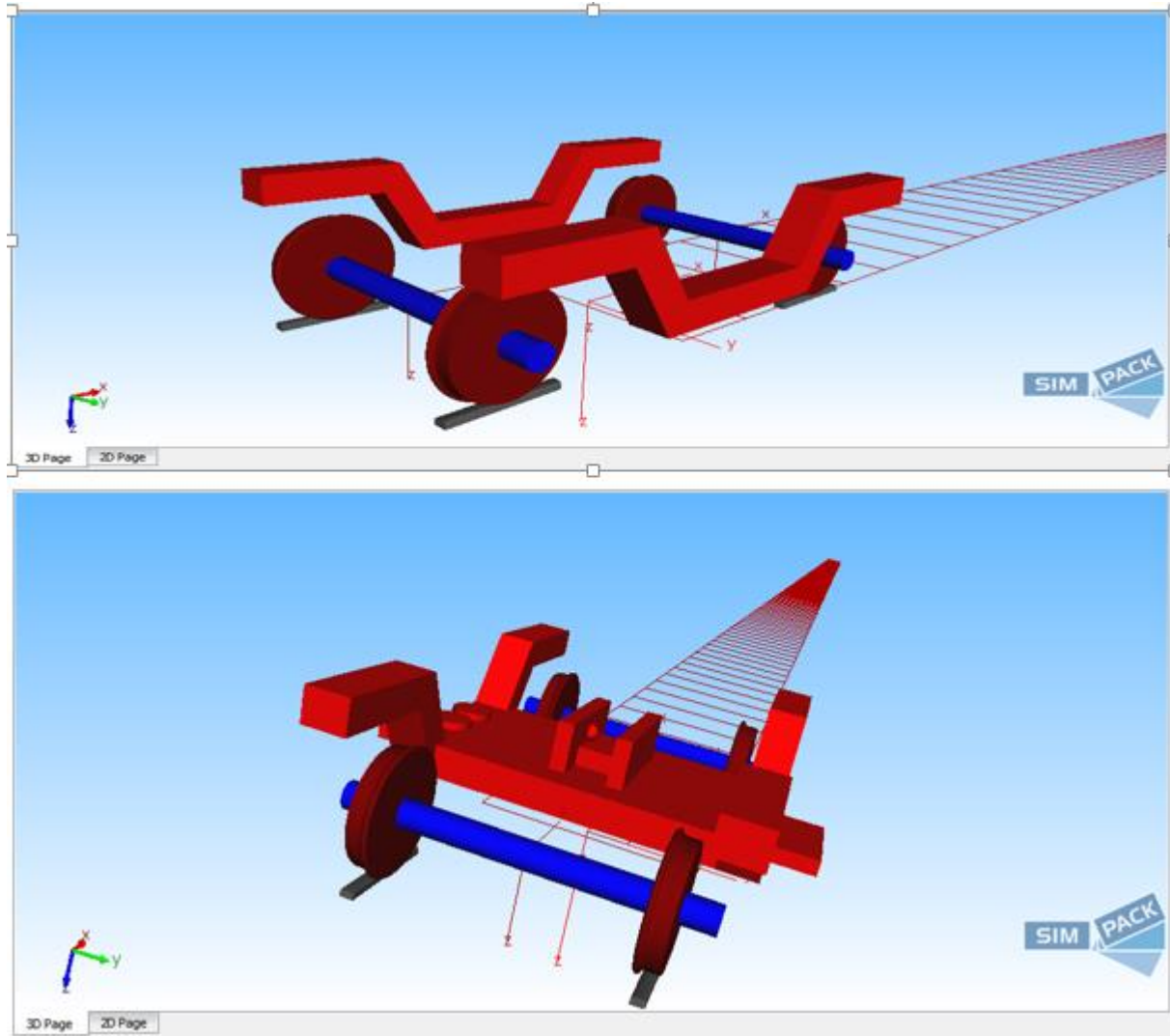


Figure:3.8: Wheelsets Pairs with Bogie frame

Bolster Frame

Create a fourth body and rename as \$B_Bolster. Fill Mass and inertia properties of the Bogie Frame assumed including all material and accessories on the bolster. Because the Bolster fixed to the car body, so take the mass near zero. $\$_{Bolster_Mass} = 1e-6$, Inertia $\$_{Bolster_Ixx} = 1e-6$, $\$_{Bolster_Izz} = 1e-6$ kgm², $\$_{Bolster_Iyy} = 1e-6$ kgm². Geometry Primitive name to '\$P_Bolster_Dummy_Top', Type '1: Cuboid', Length x=1, Length y=3, Length Z=0.2

All other Bolster Geometry Primitives are developed with Cuboid type with appropriate dimension and position.

Bolster Frame Joint: The bolster is fixed with the car body, so its movement is zero in all direction. It has zero degree of freedom. Joint type of the Bolster – 0:0 Degree of Freedom, type 0 and leave all parameter as it is or set zero and set Rotation sequence al-be-ga

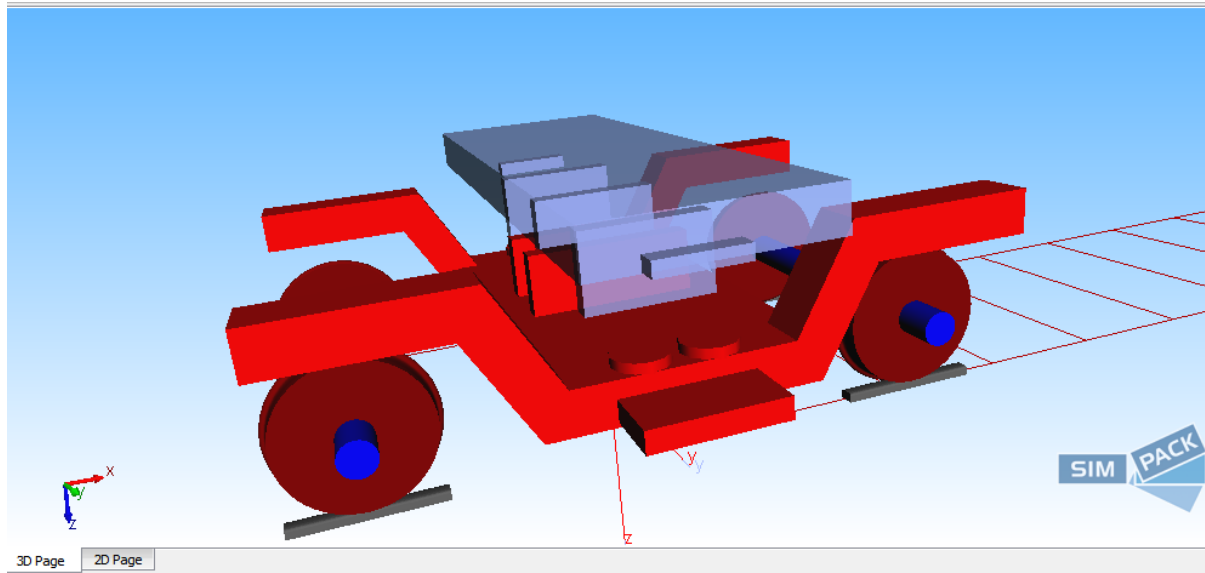


Figure: 3.9: Wheelsets Pairs with Bogie frame and Bolster

Adding a Primary Suspension system

In SIMPACK the primary and secondary suspension perform as the connection between the wheelsets (axle box) and the bogie frames and bogie frame and Bolster or car body frame respectively. The vibrations and forces coming from the wheel–rail contact are absorbed by the force element.

Primary Suspension

The primary suspensions consist of :

- Four linear parallel springs per bogie (Type: 86 : Spring-Damper ser/par cmp) which is universal force element applies spring and damper forces and torques between two markers in multiple axis directions. They can include a spring and a damper in parallel as well as another spring and damper, connected in series, parallel to these.

- Four Point to point Damper per bogie (Type: 6: Spring –Damper serial ptp) which is linear force element which connects a spring and damper in series oriented on the vertical directions as can be seen in Figure: 3.11:

To add primary suspension first create Wheelsets Markers (To Marker) and Bogie Frame Markers(From Marker) at appropriate position on Wheelset body Markers and Bogie frame body Markers with marker type:1: Identity Matrix for each element. Next create a new Force Element type: 86: for spring and Type: 6: for damper and fill the parameter value (Stiffness, Damping and Nominal Length of spring) by writing SubVar name. Change the From Marker and To Marker to above Markers and click OK. For the other three spring force element and three damper force element copy the first spring force element and damper force element then past three times for each. Next renames and changes From Markers and To Markers to the appropriate Markers.

Adding Secondary Suspenssion

The secondary suspensions consists of,

- Four linear parallel shear spring per bogie (Type: 79 : Shear-Spring cmp), a typical coil spring (helical spring) with shear forces and bending torques coupled.
- In vertical direction, two linear Point to point Damper per bogie, having linear spring-damper element in series (Type:6:Spring –Damper serial ptp),
- In lateral direction; two linear Point to point Damper per bogie, having linear spring-damper element in series (Type:6:Spring –Damper serial ptp),
- In lateral direction; two bumpstops in each bogie; represented by non-linear spring with damping Type:5: Spring –Damper serial cmp,
- In roll direction; one antiroll bar per bogie with universal force element applies spring and damper torques between two markers in the three rotation directions. Type:13: Spring_Damp Rot meas Inp cmp.
- In longitudinal direction; one traction rod in the middle of each bogie, represented by point to point linear spring element defined by length of un-stretched length, linear stiffness of spring and additional pre load Type: 1: Linear Spring ptp as can be seen in Figure:3.11 ,
- In yaw direction; one linear yaw damper at each bogie side of the bogie, having linear spring-damper elements in series.

To add secondary suspension first create Bogie Frame Markers (To Marker) and Bolster Markers(From Marker) at appropriate position on Bolster body markers and Bogie frame body Markers with maker type:1: Identity Matrix. Next create a new Force Element type: Type: 79: for spring, Type: 6: for damper (for Traction rod, lateral and verticl). Type:5: for bump stop, Type: 1: for Traction rod spring and Type:13: for antiroll bar and fill the parameter value (Stiffness, Damping and Nominal Length of spring) by writing SubVar name. Change the From Marker and To Marker to above Markers and click OK. For the other three spring force element and one vertical damper force element and one Lateral damper force element copy the first spring force element and damper force element then past the spring force element three times and the damper force element one time for each. Next renames and changes From Markers and To Markers to the appropriate Markers.

Adding a Primary and Secondary Suspension Force Element Primitive

To create Force Element Primitive first On the Isys Reference System ('\$B_Isys') Markers, create a new Marker of type 80, 'point-point' for each element and change From Markers and To Markers to appropriate or as the same to force element parts for each elements.

Next create Force Element Primitive On the Isys Reference System ('\$B_Isys') Geometry, create new primitive for each elements as follow:

- For Primary Spring and Secondary Spring
 - Primitive type: 13: Spring
 - Enter appropriate Number of winding, Spring length and Diameter
 - Reference Marker: from the Isys Reference System ('\$B_Isys') Markers change to appropriate point-to point maker early created.
- For Primary vertical Damper and Secondary Vertical and Lateral Damper
 - Primitive type: 2: Cylinder
 - Enter appropriate Number of planes, Height, Outer Diameter, Z-position and Angle(alpha)
 - Reference Marker: from the Isys Reference System ('\$B_Isys') Markers change to appropriate point-to point maker early created.
- For Primary vertical Damper and Secondary Vertical and Lateral Damper Rotated body
 - Primitive type: 7: Rotational body

- Enter appropriate Number of planes, Points(X2(Length), Y1(upper diameter) and Y2(lower diameter)) and Angle(beta)
- Reference Marker: from the Isys Reference System ('\$B_Isys') Markers change to appropriate point-to-point marker early created.

After adding primary and Secondary suspension system the Bogie model will like the next figure:3.11

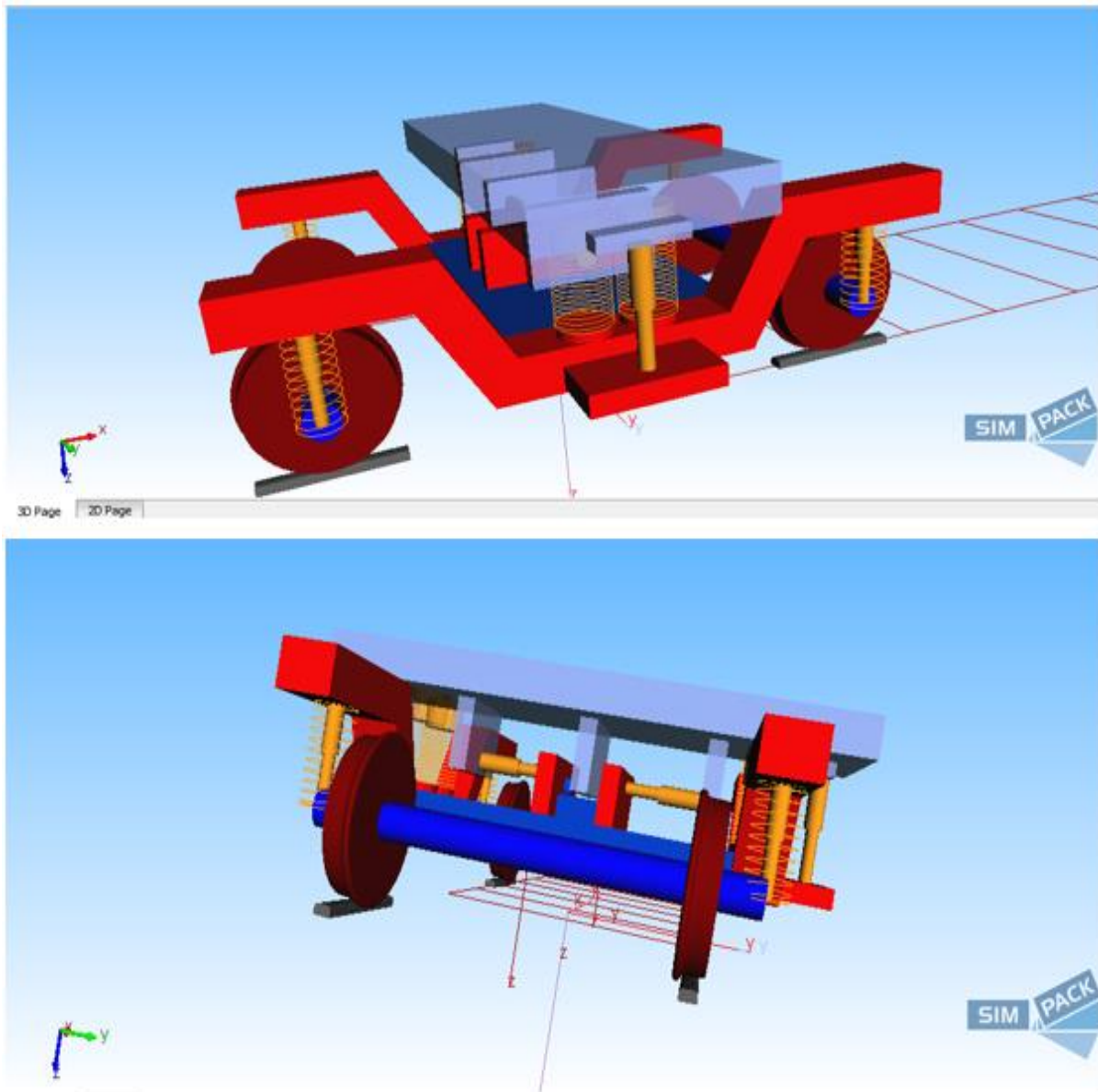


Figure:3.11: Bogie with Force element

3.3.1.2 Development of Complete Vehicle

In this study the model created for the simulations is composed by the following bodies, see Table:3.2:. It should be noted that all bodies are considered rigid bodies.

Body Name	Quantity of body	Total degree of Freedom
Wheelsets	4	4x6=24
Bogie Frame	2	2x6=12
Car Body	1	1x5=6
Total Degree of freedom		41DOFs

Table:3.1: Model Vehicle Composition and Degree of freedom

To create the complete railway vehicle first create new folder rename 'Thesis model' and open the folder and create othe a new folder and rename 'Model'. Next the Bogie model folder 'Bogie Database' folder put in 'Thesis model' folder and create new SIMPACK model using the 'General' template and save→ Thesis mode→ Model.

Create Car body

First assign all input parameter to SubVar and then rename the body name as \$B_Carbody or other name with \$B_. Fill mass and inertia properties of the Carbody: \$ _Carbody_Mass =32000 kg, Inertia \$ _Carbody_ $I_{yy} = I_{zz} = 19700000 \text{ kgm}^2$, \$ _Carbody_ $I_{xx} = 56800 \text{ kgm}^2$. Primitive name to ' \$P_Carybody_Car_Body', Type:21 'wheel rail Cab', Prameter Leyer Length L9 in X = \$ _Bogie_base=19, Angl alpha=180deg and other primitive parameter set as appropriate.

Car Body Joint: Joint type: 9: Rheonomic rail track joint. This joint prescribes the movement of a rail vehicle component along a track in space. Other directions than the movement along the track are free. This Joint is typically used for one carbody in a model or for a 'traction body' that resembles a locomotive for pulling a pushing a train. All state leave as it is or zero position and in parameteric Icon:

- Operating mode: change constant velocity to U_vector input $s(t)$
- Degree of freedom: 5(Y,Z,phi,psi,gamma)
- U-vector ID $s(t)$: \$UE_T_Speed profile_Value (The u-Vector Element for the position along the Track. This setting is required.)
- U-vector ID $sd(t)$: \$UE_T_Speed profile_1st_deriv (The u-Vector Element for the velocity along the Track. This setting is required.)
- U-vector ID $sdd(t)$: \$UE_T_Speed profile_2nd_driv (The u-Vector Element for the acceleration along the Track. This setting is required.)
- Longitud. Position offset: \$ _CarWagon (A constant offset that is added to the s value read from the u-Vector Element in parameter 16)

Define a standard straight and Curved Track

Track definition in SIMPACK

The Cartographic Track assembles the track line from so-called segments in the three directions 'horizontal', 'superelevation' and 'vertical'. The directions are handled independently, i.e. the segments in the different directions do not have to have the same lengths. The segments are stringed together in the sequence defined by the user.

For this Study the truck scenarios are:

Straight track scenario

Initial straight track scenario is having a 500 meters long track and end straight track scenario is having a 600 meters long track.

Curved track scenario

The curved track scenario is having a 570 meters long track specify as follow:

- Initial transition curve segment length 190m and radius 380m
- Curved segment length 190m and radius 380m
- End transition curve segment length 190m and radius 380m
- The half-length of the smoothing section from the current to the next segment, $L_{smo}/2=1.5m$

Create Track: Track element: \$Trk_Track, Type: Rail

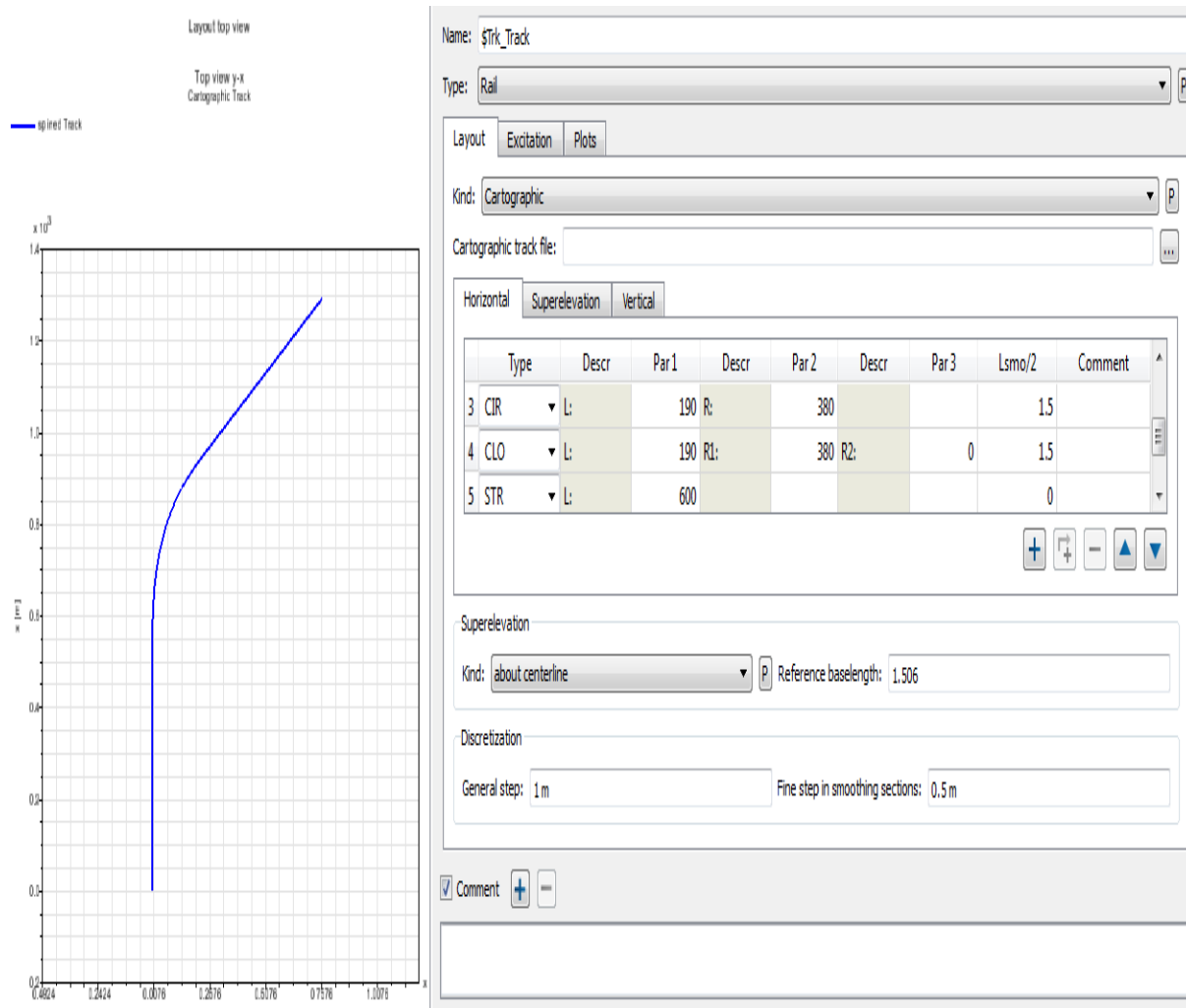


Figure: 3.13: Standard Horizontal Straight and Curved Track Layout

Create Input Function, U-vector, Excitation and Sensor

- Create Input function for Speed profile of the Track
- Create U_vector speed profile(value, 1st driv and 2nd driv)
- Create Excitation type:20: Speed profile along Track
- Create different sensors

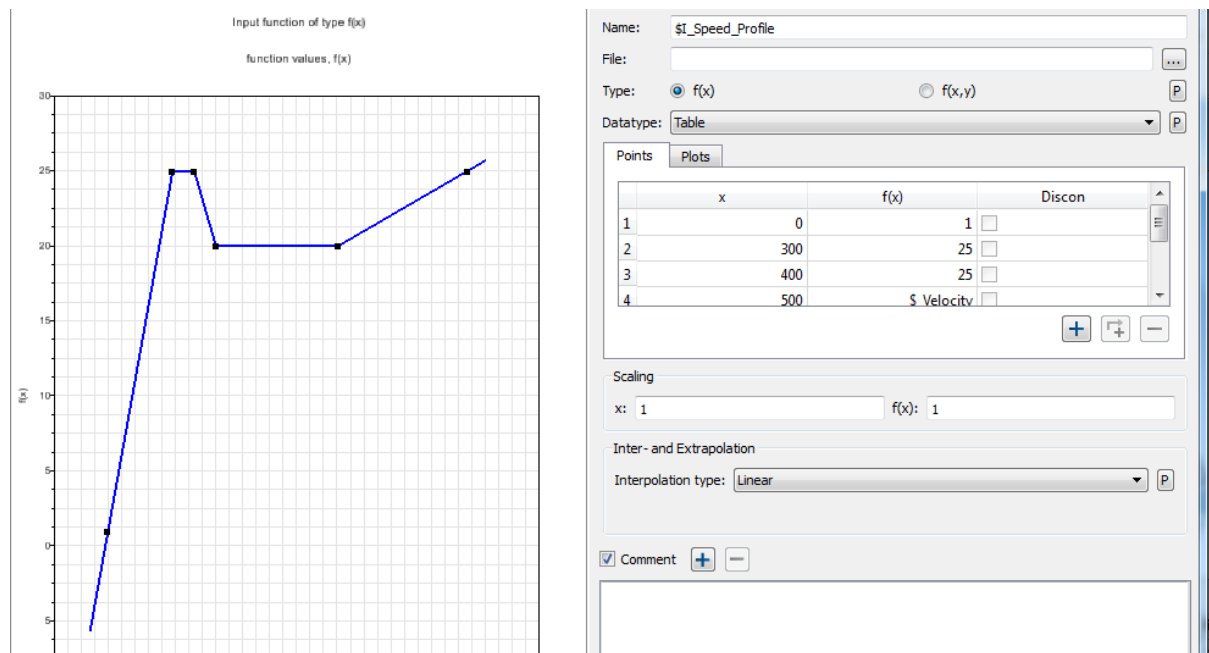


Figure:3.14: Speed Profile along Straight and curved Track

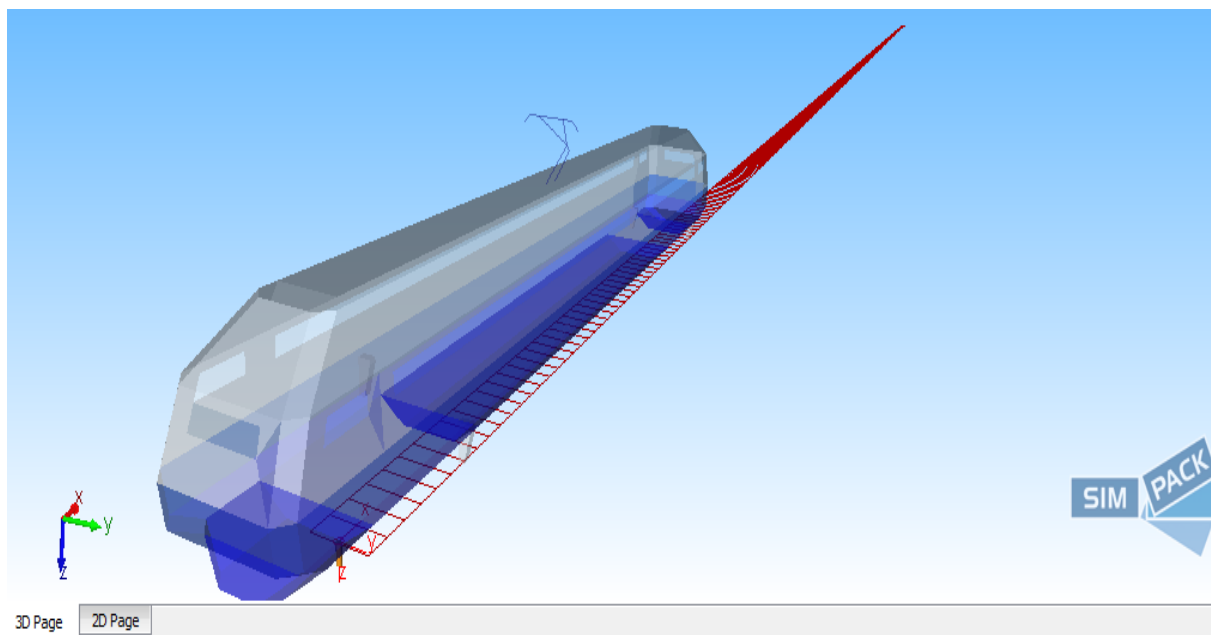


Figure:3.15: Tarack and Wheel Rail Cab

Adding Substructures (Front and rear Bogie) to the existing model

First add searches path entities in global setting the folder Bogie Database by opening Search path.

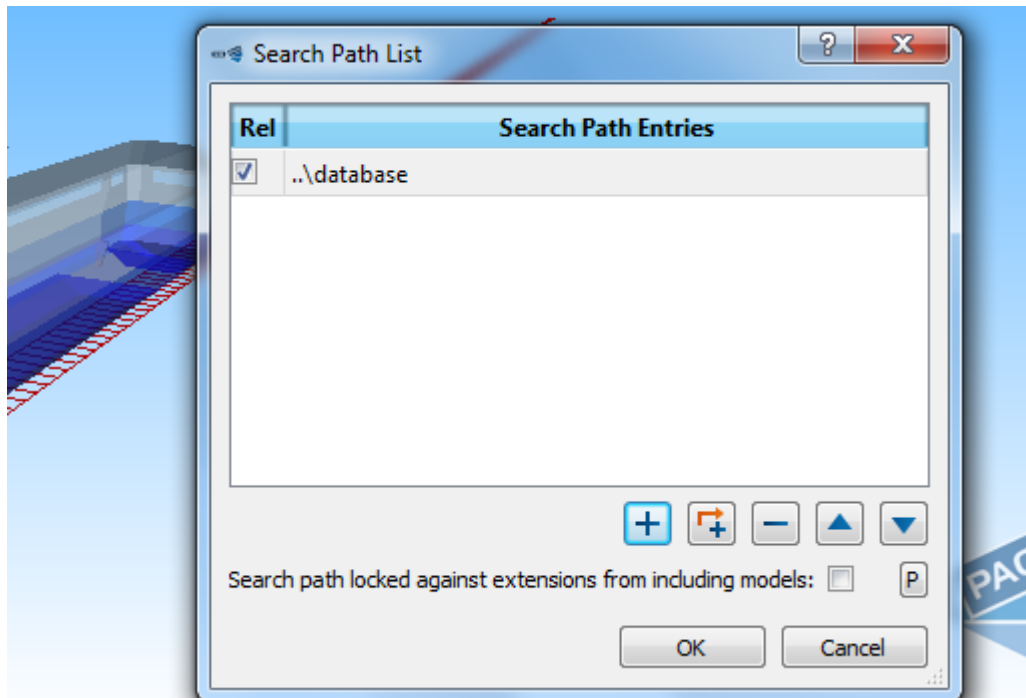


Figure:3.16: Search path Entries Dialog Box

Next create Substructures and rename `$S_Front_Bogie` click Ok. Next Substructure of `$S_Front_bogie` appear (Substructure ID:1), then search model filename 'Bogie' and click Ok. Add second Substructure and rename `$S_Rear_Bogie`(Substructure ID:2) and follow the same steps.

Reset the Front Bogie and Rear Substructure Joint position as follow:

Front Bogie Substructure

- `$J_Frame`: Longitudinal position: `$_Front_Bogie_Fx`
- `$J_WS_F`: Longitudinal position: `$_Front_Bogie_FWS` and Vertical position: `$_Axle_Z`
- `$J_WS_R`: Longitudinal position: `$_Front_Bogie_RWS` and Vertical position: `$_Axle_Z`
- `$J_Bolster_Dummy`: as it is or all parameter set zero

Rear Bogie Substructure

- `$J_Frame`: Longitudinal position: `$_Rear_Bogie_Fx`
- `$J_WS_F`: Longitudinal position: `$_Rear_Bogie_FWS` and Vertical position: `$_Axle_Z`
- `$J_WS_R`: Longitudinal position: `$_Rear_Bogie_RWS` and Vertical position: `$_Axle_Z`
- `$J_Bolster_Dummy`: as it is or all parameter set zero

Therefore the complete railway vehicle shown in Figure: 3.17.

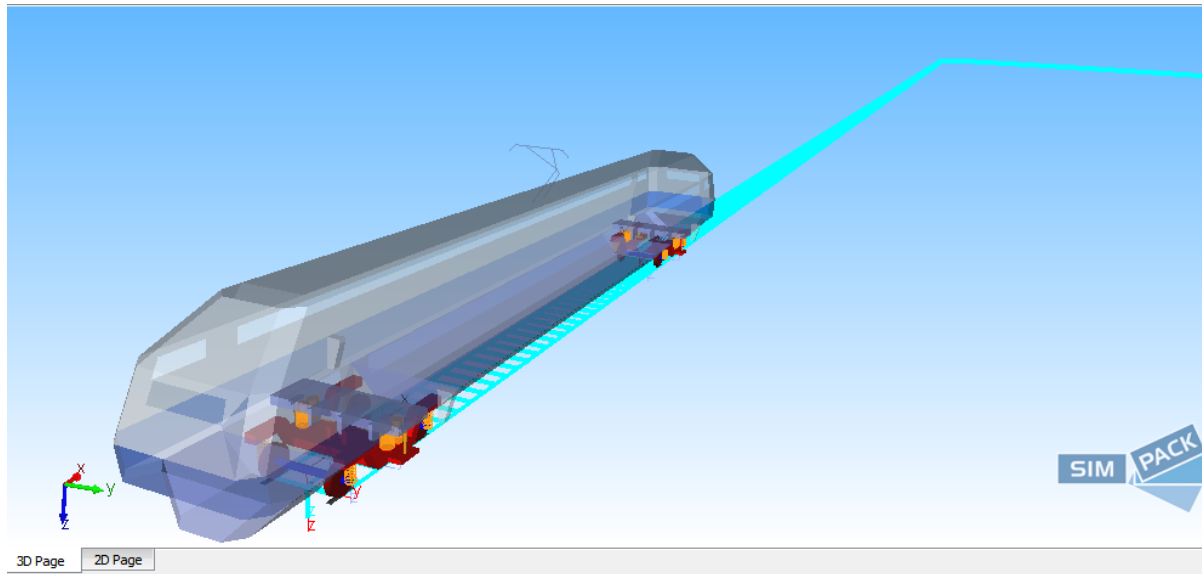


Figure: 3.17 Complete Railway Vehicle Model

3.3.2 Static equilibrium and preload calculation

After the railway vehicle model along with the track scenario created, it is time to prepare it for the simulations. Before simulation, a static equilibrium position must be determined. In SIMPACK there are two ways to bring a model into equilibrium, the static equilibrium and the preload solver. For rail vehicles, the Preload solver is mostly used. It calculates the value of the preload of every suspension element taking charge of the effect of the gravity on different masses. The Preload solver calculates the preloads automatically so that the model is in an equilibrium, i.e. the residual accelerations become zero or at least very small. For the illustrated case, this value is smaller than $3.15 \times 10^{-5} \text{ m/s}^2$ shown in Appendix: figure: B.1 so can be said that the effect of the gravity is absorbed by the force elements.

After the force elements are in equilibrium, determine the equilibrium position of the full model in the three directions. SIMPACK provides the tool “Static Equilibrium” which calculates the static position of each body and gives the “Maximum force residuum” in that position. This value, as in the previous case for the “Preload calculation”, must be equal to zero or very small. For the illustrated case, the static equilibrium shown in Appendix: figure: B.2:

Finally, once SIMPACK finds a suitable equilibrium position, the model is ready to be used in the simulations.

3.3.3 “Solver Settings” panel: time integration off-line

The Time Integration solver determines the complete behavior of the non-linear model in time, by solving the full set of non-linear equations of motion over the time. The solver uses one of various multi-purpose and special-purpose integrators, which can be selected in the Solver Settings. During a Time Integration, the system’s equations of motion are solved for a specified period, resulting in time histories for all states. Solver parameters, such as solver type, initial and final time, step-size, error tolerances, etc., will be set for the offline Time Integration. Set end time: 100sec

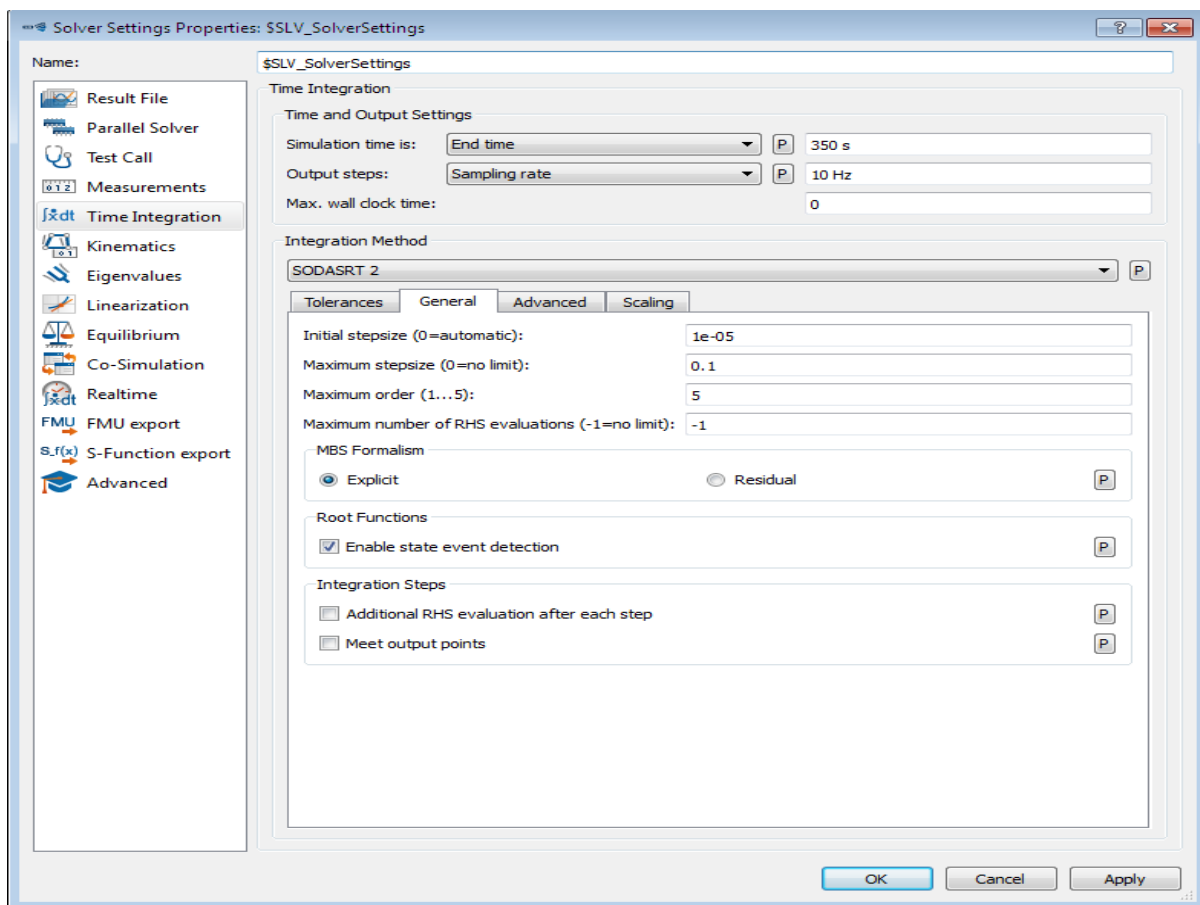


Figure:3.18: Solver Settings Dialog Box

3.3.4 Post Processing

The post-processing step deals with the result plotting and extracted data from the simulation. The SIMPACK post process program generate detailed results for each body forming the multi-body

model in the railway vehicle. Based on this procedure important rail vehicle performance will be simulated, plotted and animated. For this study the result of derailment coefficient (Y/Q) and Y sum of lateral rail force extracted and plotted.

Finally based on the extracted data, evaluation and discussion to the objective function of the study performed.

3.3.5 Expected outcome

After the analysis performed expected that differentiate which railway parameter variation have risk of derailment, which parameters variation have safe to risk of derailment and which parameter have no effect to risk of derailment. As the same time the parameter increase or decrease to outcome risk of derailment or safe to risk of derailment. In addition, at the parameters variation outcome safe to derailment how much the value of critical speed improved with in the given variation interval?

Chapter Four

4. Simulation Result and Evaluation of Effect of Railway Vehicle Parameter Variation on Derailment

To simulate the objective function, first simulate the model by using SIMPACK experiment with original/design parametric value to fix the curved critical speed of the rail vehicle for the given curved track scenario. Afterward Derailment coefficient and Y sum lateral rail force of the rail vehicle obtained from simulation by varying each railway vehicle parameters. The result from SIMPACK post-process present in 2D plotted.

4.1 Determination of Curved Critical Speed of the model

On straight track scenario, the maximum vehicle speed is set to be 80Km/h. The maximum admissible speed of the vehicle comes from the standard EN-14636 in which it is defined that the speed in straight track for the safety assessment must be equal to $1.1 \times \text{Maximum speed} = 1.1 \times 80 = 88 \text{Km/h} = 25 \text{m/s}$.

For this study on the straight track the maximum allowable speed has taken 25 m/s. But for curved track scenario (radius (380m) the maximum allowable speed determined with SIMPACK simulation experiment. SIMPACK DoE ('Design of Experiments') allows easy configuration and control of experiments on existing SIMPACK models. The module itself and its terminology follow the common rules of factorial experiment design.

The experiment done on SIMPACK by varing the speed on curved track start from maximum alloable speed on straight track decrease with 1m/s. The maximum speed at which not derailment of the vehicle appear taken as Maximum allowable speed of the vehicle on curved track.

Then the speed profile input set again with determined Maximum allowable speed of the vehicle on curved track. At this speed the railway vehicle move without deraiment along the curved track. Result of SIMPACK DOE expriment for maximum alloable speed on curved track present below:

The user defines in this View Set the DoE responses, i.e. the results element from the model that shall serve to assess the model's behavior. To keep the amount of data low, it is recommended that

only a very limited set of responses be selected. For this purpose front bogie front wheelset derailment coefficient taken to see whether the vehicle derail or not.

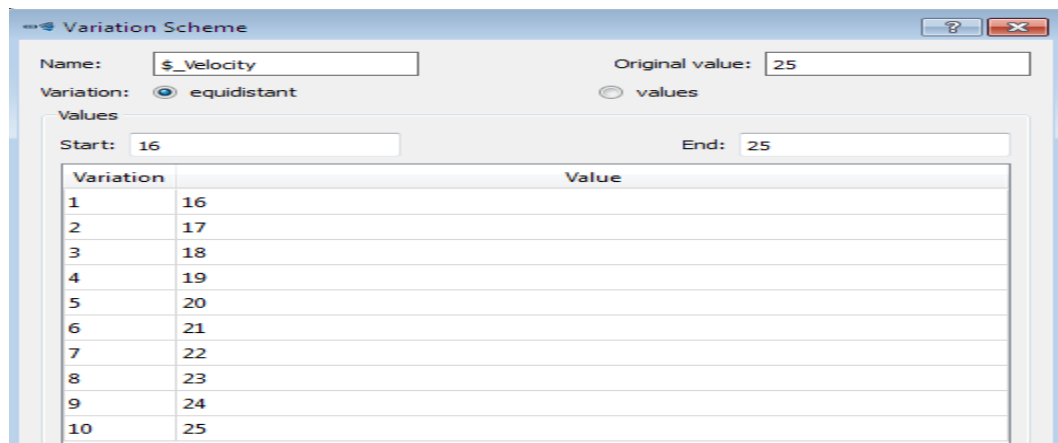


Figure: 4.1: Speed variation 16m/s to 25m/s

Experiment status: **not running**

Run Id	Start Time	End Time	Model Time	Progress
1	25 8:15 2015 20:2...	25 8:15 2015 20:2...	100 s	100 %
2	25 8:15 2015 20:2...	25 8:15 2015 20:2...	100 s	100 %
3	25 8:15 2015 20:2...	25 8:15 2015 20:2...	100 s	100 %
4	25 8:15 2015 20:2...	25 8:15 2015 20:2...	100 s	100 %
5	25 8:15 2015 20:2...	25 8:15 2015 20:2...	100 s	100 %
6	25 8:15 2015 20:2...	25 8:15 2015 20:2...	67.6 s	100 %
7	25 8:15 2015 20:2...	25 8:15 2015 20:2...	65.7 s	100 %
8	25 8:15 2015 20:2...	25 8:15 2015 20:2...	64.5 s	100 %
9	25 8:15 2015 20:2...	25 8:15 2015 20:2...	63.7 s	100 %
10	25 8:15 2015 20:2...	25 8:15 2015 20:2...	63.1 s	100 %

Figure: 4.2: Speed Experiment result

Experiment Result

Take two results from Figure: 4.2 No five and six. Result No 5 (at $V=20\text{m/s}$) complete integration time, but result No 6 (at $V=21\text{m/s}$) not move longer than 67.6s. And farther experiment results the maximum allowable speed (curved critical speed) for given curved track seniors is 20.7m/s .

Post Processing

SIMPACK Post allows you to visualize practically any result data that you are interested in, both as 2D and 3D plots as well as in a 3D animation. From the above experiment, three result selected and plotted.

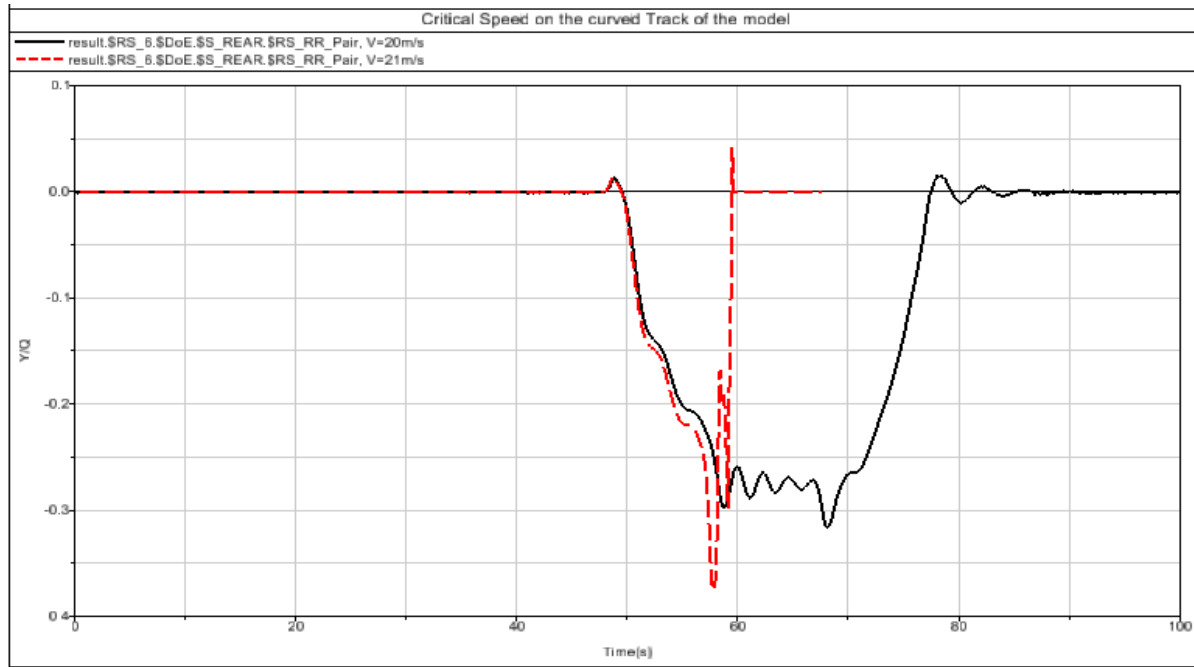


Figure: 4.6: Derailment Coefficient

Therefore the speed profile will reset with determined maximum allowable speed on curved track as shown below.

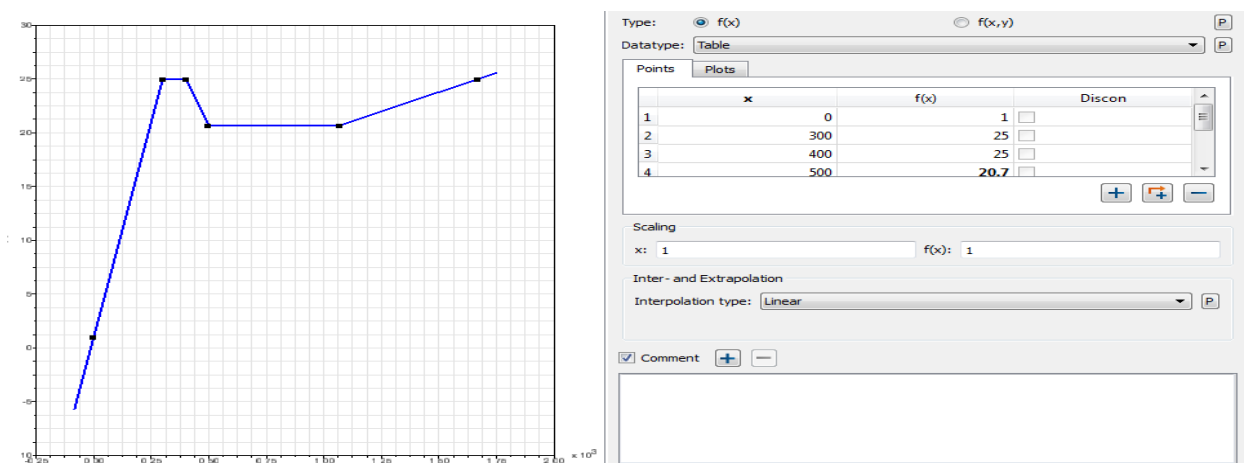


Figure: 4.3: Speed Profile

4.2 Evaluation of Effect of Railway Vehicle Parameters Variation on Derailment

For the purpose of this study the rail vehicle parameters were vary from -25% to +25% of their original values and the parameters are:

N o	Name of Parameters	Original Value	(Original-25%) Value	(Original+25%)Value
1	Mass of car body	32,000Kg	24,000Kg	40,000Kg
2	Carbody rolling mass moment of inertia(Ixx)	56,800 Kg/m ²	42,600 Kg/m ²	71,000 Kg/m ²
3	Carbody pitch mass moment of inertia(Iyy)	1.97 x 10 ⁷ Kg/m ²	1.4775 x10 ⁷ Kg/m ²	2.4625 x 10 ⁷ Kg/m ²
4	Carbody yaw mass moment of inertia(Izz)	1.97 x 10 ⁷ Kg/m ²	1.4775 x10 ⁷ Kg/m ²	2.4625 x 10 ⁷ Kg/m ²
5	Vertical mass center of Carbody	-1.8 m	-1.35 m	-2.25 m
6	Secondary spring longitudinal shear stiffness	80,000 N/m	60,000 N/m	100,000 N/m
7	Ssecondary spring lateral shear stiffness	430, 000 N/m	322,500 N/m	537,500 N/m
8	Secondary spring Vertical stiffness	6.0 x 10 ⁶ Ns/m	4.5 x 10 ⁶ Ns/m	7.5 x 10 ⁶ Ns/m
9	Secondary damper lateral damping	6.0 x 10 ⁶ Ns/m	4.5 x 10 ⁶ Ns/m	7.5 x 10 ⁶ Ns/m
10	Wheel base – center to center distance of wheel sets in a bogie	2.23 m	1.6725 m	2.7875 m
11	Wheel Diameter	0.33 m	0.305 m	0.355 m

Figure:4.1:Values of Parameter Variation

4.2.1 Effect of Car body mass

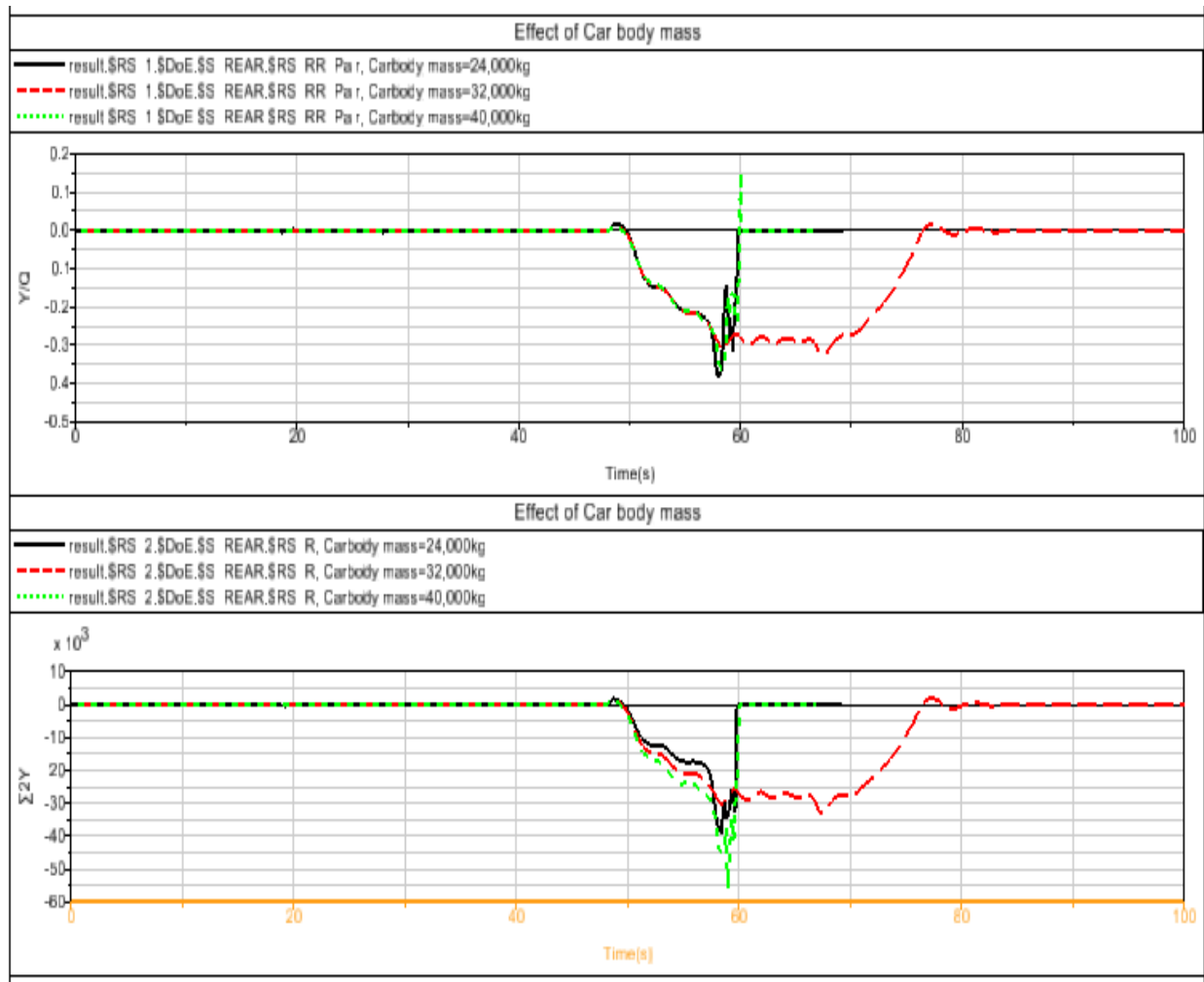


Figure: 4.4: Effect of Carbody mass

In figure: 4.4 show that both decreasing and increasing carbody mass lead to vehicle derailment. Further experiment Figure: 4.5 show that the derailment coefficient is decrease with increasing Carbody_Mass, but Y sum of lateral rail force is increases. Therefore increasing carbody mass increase the lateral force b/n rail and wheel, so lead to track irregularities and the mass increase more from original value derailment is occur. On the other hand the derailment coefficient is increase with decreasing Carbody_Mass from original value, but Y sum of lateral rail force is decrease. Therefore more decreasing carbody mass due to wheel climbing lead to derailment of vehicle. From more further experiment -5.25% to +15.75% of original mass safe to risk of deraiment at curved critical speed, but decreasing below and increasing above this percentage vehicle derailment occur.

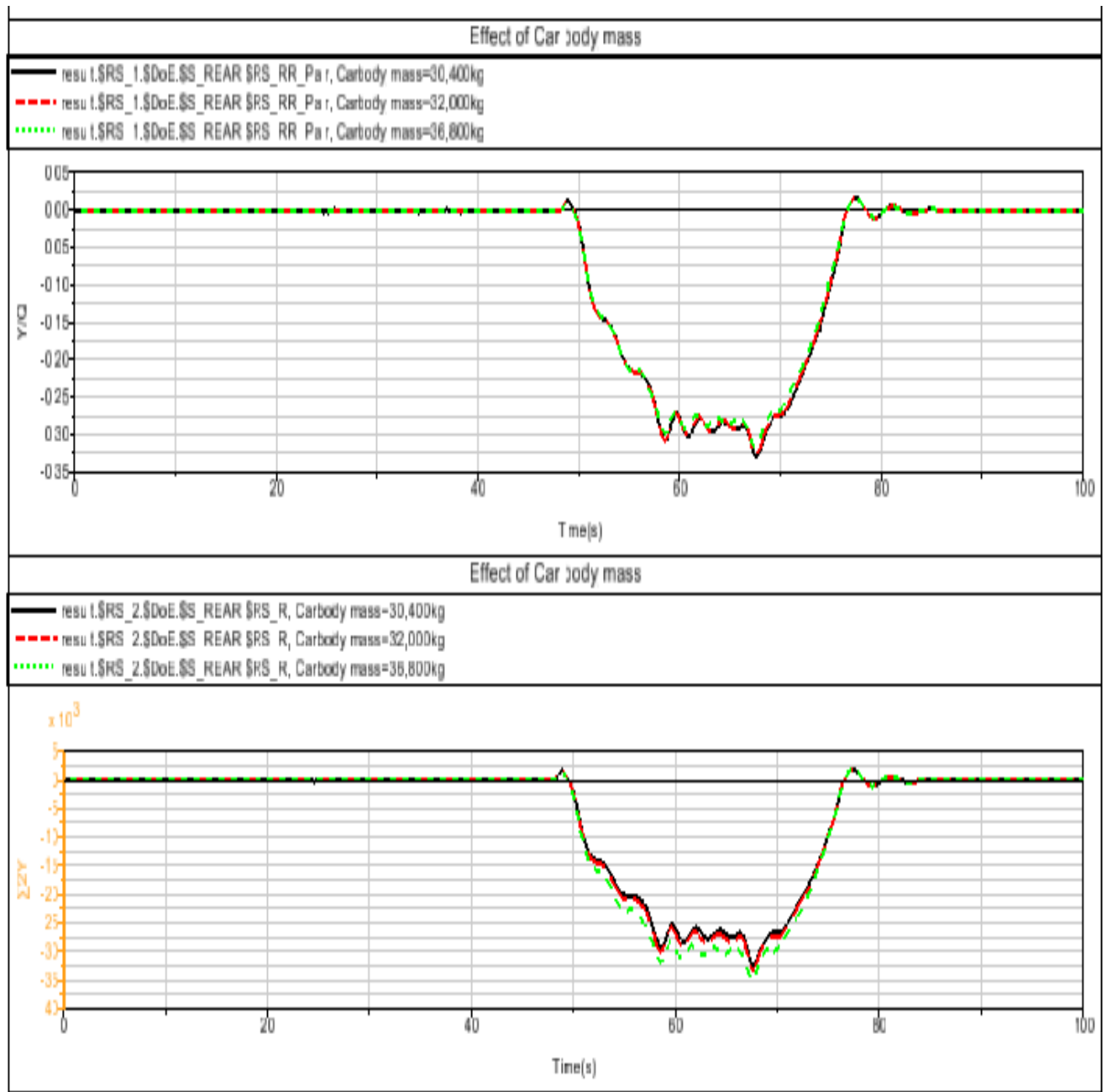


Figure: 4.5: Effect of Carbody mass further experiment

4.2.2 Effect of Mass Moment of Inertia of Carbody

4.2.2.1 Effect of Roll mass moment of inertia of Carbody (I_{xx}) of car body

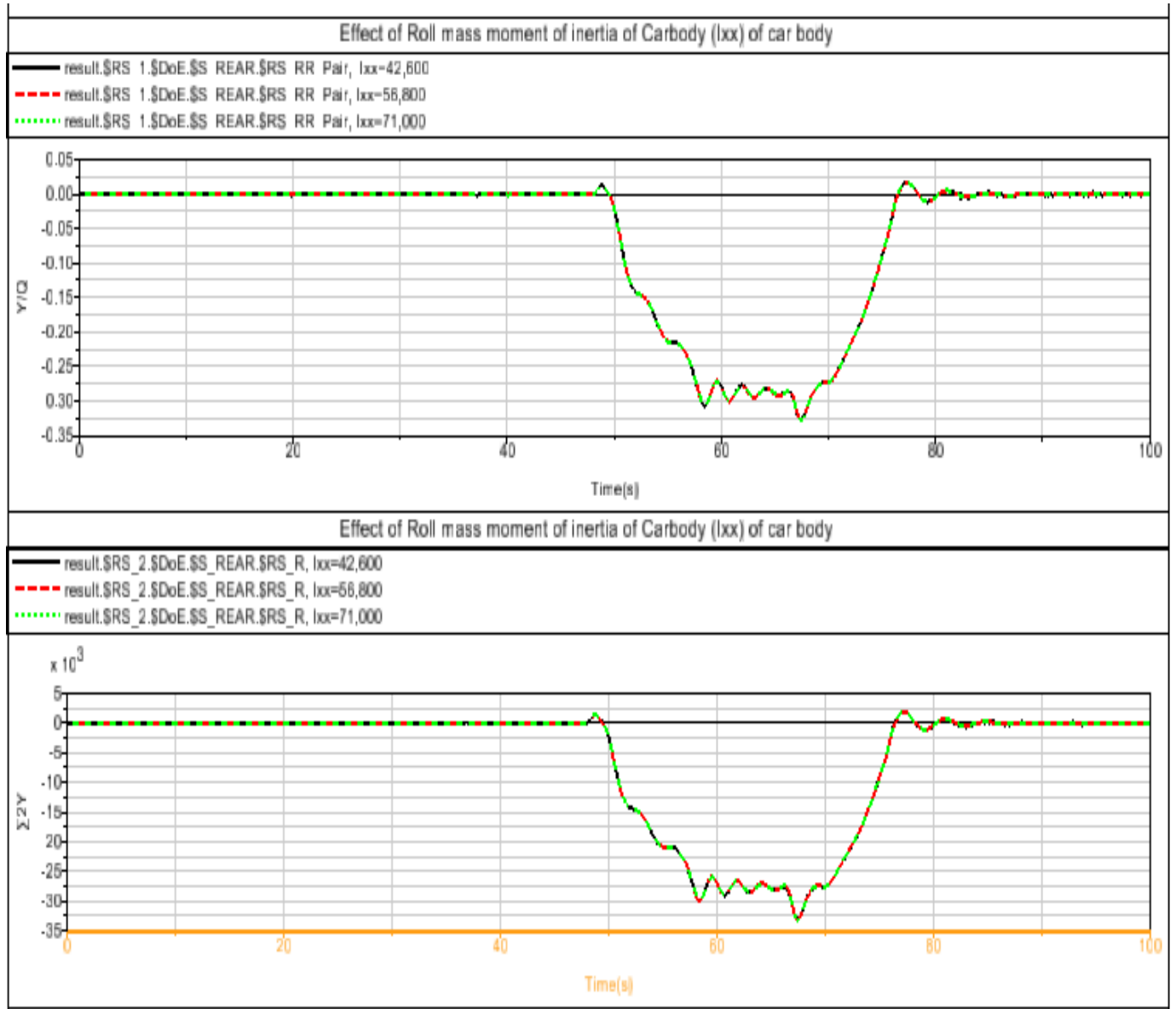


Figure: 4.6: Effect of Roll mass moment of inertia of Carbody (I_{xx}) of car body

In figure:4.6 both derailment coefficient and Y sum lateral rail force have no any change. Therefore, increase or decrease of longitudinal mass moment of inertias(I_{xx}) of carboy have no effect on derailment when vehicle running on curved track.

4.2.2.2 Effect of Pitch mass moment of inertia of Carbody (I_{yy}) of car body

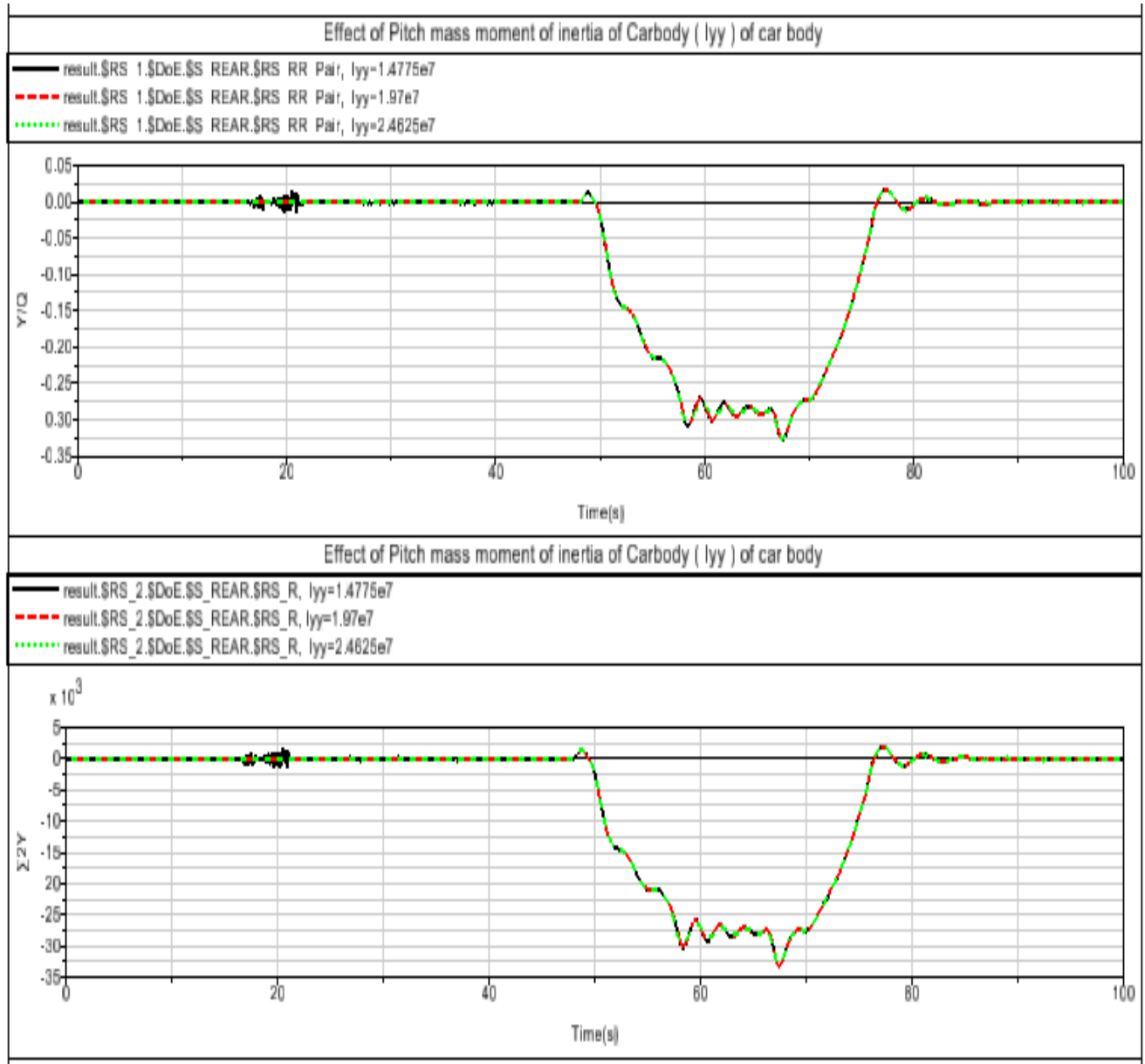


Figure: 4.7: Effect of Pitch mass moment of inertia of Carbody (I_{yy}) of car body

In figure:4.7 show both derailment coefficient and Y sum rail lateral force have no any change. Therefore, increase or decrease of longitudinal mass moment of inertias(I_{yy}) of carboy have no effect on derailment when vehicle running on curved track.

4.2.2.3 Effect of Yaw mass moment of inertia (Izz) of Carbody

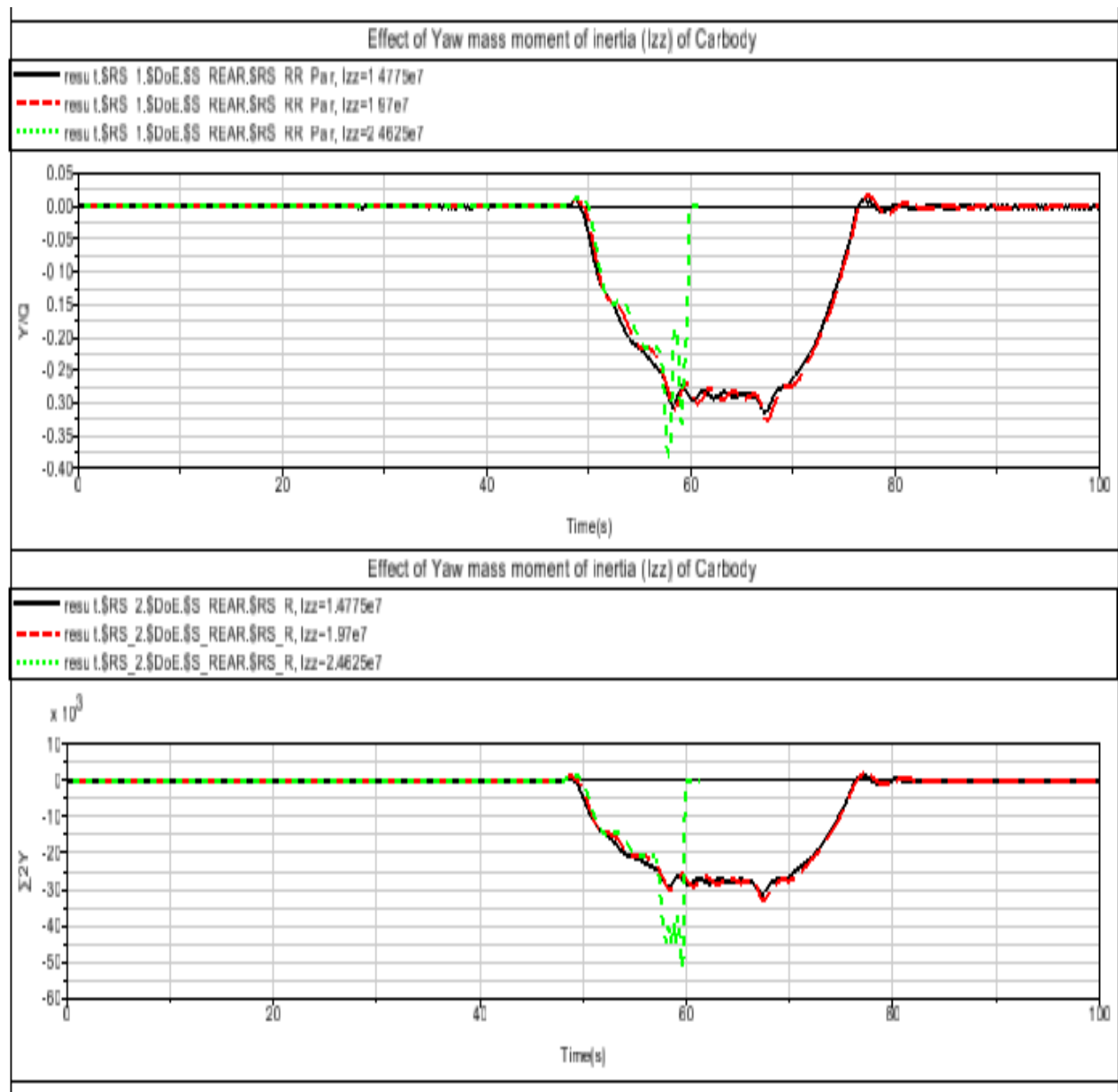


Figure: 4.8: Effect of Yaw mass moment of inertia (Izz) of Carbody of car body

In figure:4.8 both derailment coefficient and Y sum of lateral rail force increase due to increment of carbody Izz. Therefore, increment of vertical mass moment of inertias(Izz) of carboy result vehicle derailment. But decrement of vertical mass moment of inertias(Izz) of carboy result safe to derailment on vehicle running on curved track. Further experiment show that decreasing 25% from original value improve the critical speed by 1km/h and increasing above 1.5% result vehicle derailment at curved critical speed.

4.2.3 Effect of Vertical mass center of Carbody

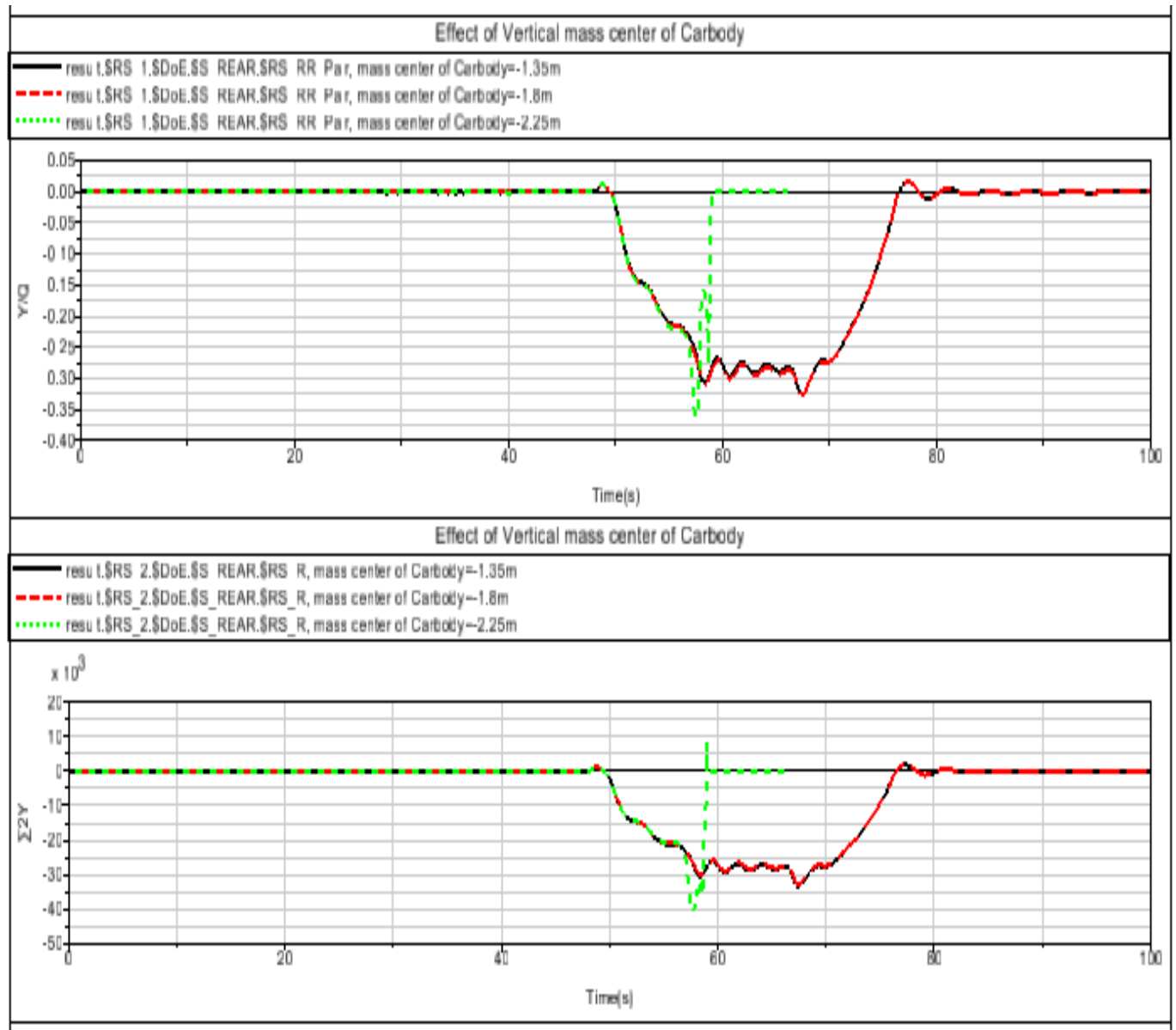


Figure: 4.9: Effect of Vertical mass center of Carbody

In figure:4.9 both derailment coefficient and Y sum of lateral rail force decrease with decreasing vertical mass center result safe to derailment and increasing vertical mass center lead to vehicle derailment. Further experiment show that decreasing 25% from original value improve the critical speed by 3.6km/h and vertical center of mass value increase above 0.875% of original value vehicle derailment occur at curved critical speed .

3.3.4 Effect of Secondary spring stiffness values

3.3.4.1 Effect of Secondary spring longitudinal shear stiffness

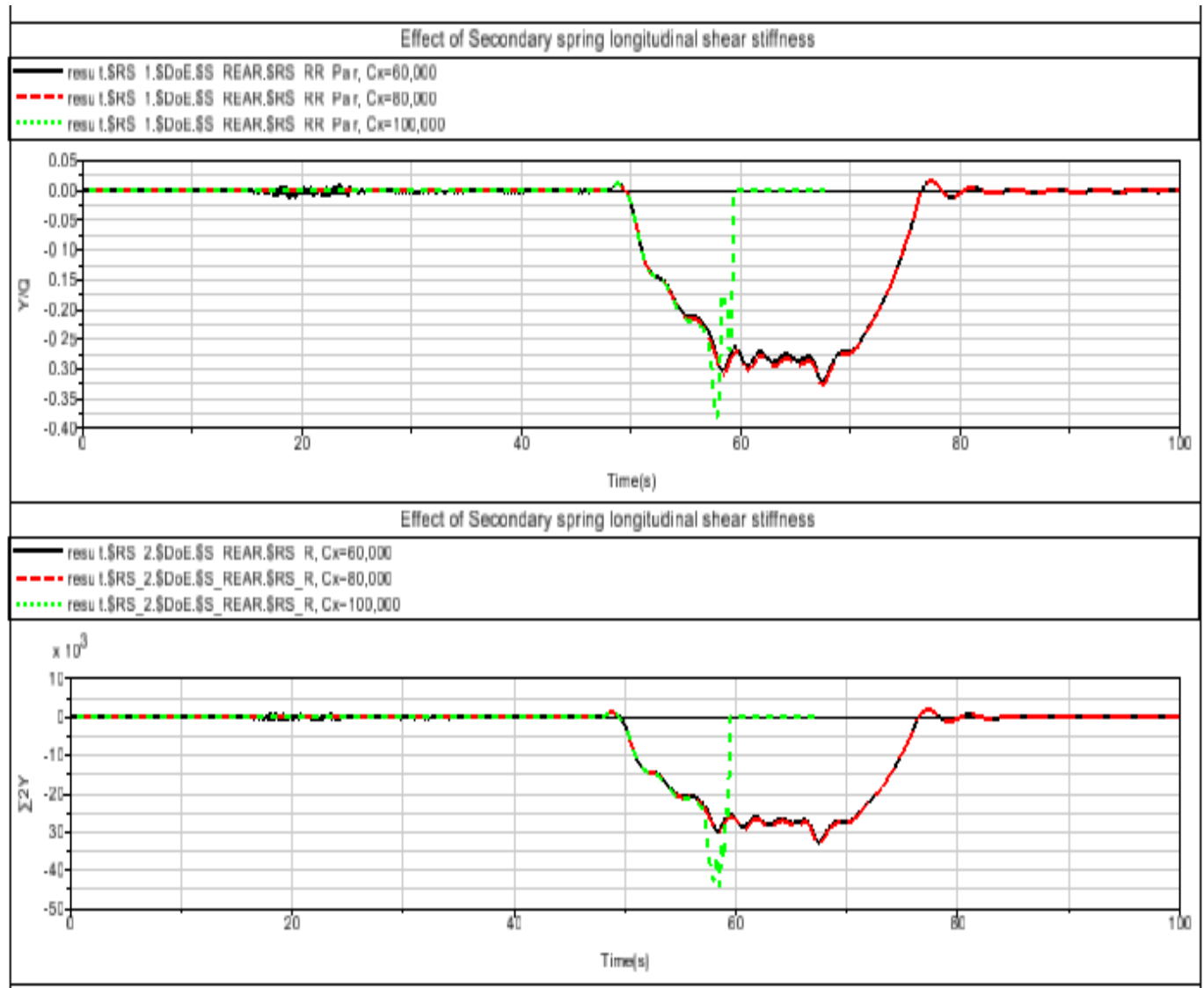


Figure: 4.10: Effect of Secondary spring longitudinal shear stiffness

In figure: 4.10 both derailment coefficient and Y sum rail lateral force increase with increasing secondary spring longitudinal shear stiffness, lead to risk of derailment. But decreament of secondary spring longitudinal shear stiffness result safe to risk of derailment on vehicle running on curved track. Further experiment show that decreasing 25% from original value improve the critical speed by 1.8km/h and increasing above 0.5% of original value lead to vehicle derailment at curved critical speed.

3.3.4.2 Effect of secondary spring lateral shear stiffness variation

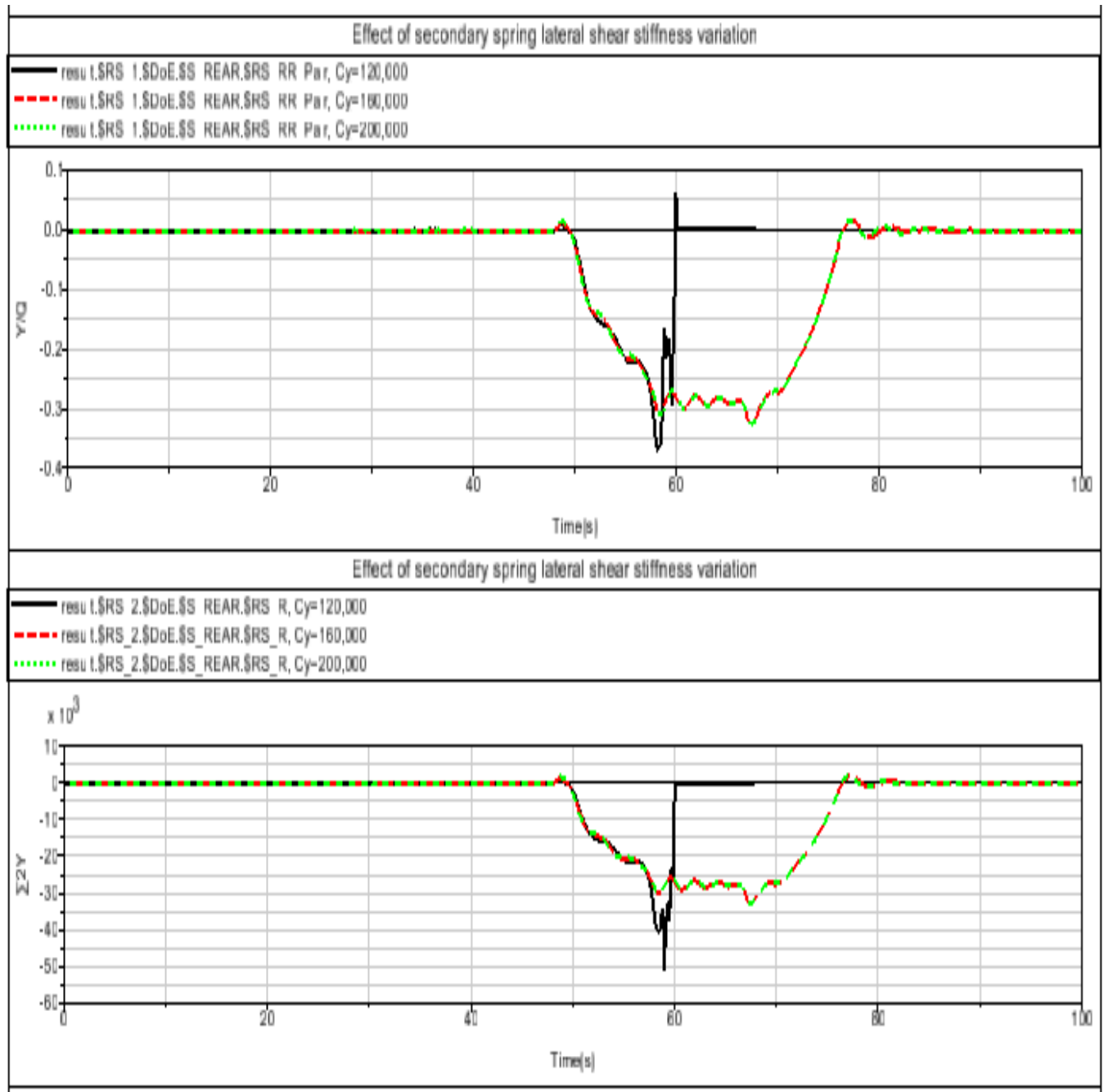


Figure: 4.11: Effect of secondary spring lateral shear stiffness variation

In figure:4.11 both derailment coefficient and Y sum of lateral rail force have no change due to increment of secondary spring lateral shear stiffness, but decreasing secondary spring lateral shear stiffness from original value result vehicle derailment. Further experiment show that decreasing above 4.875% of original value vehicle derailment occur.

3.3.4.3 Effect of secondary spring vertical stiffness variation

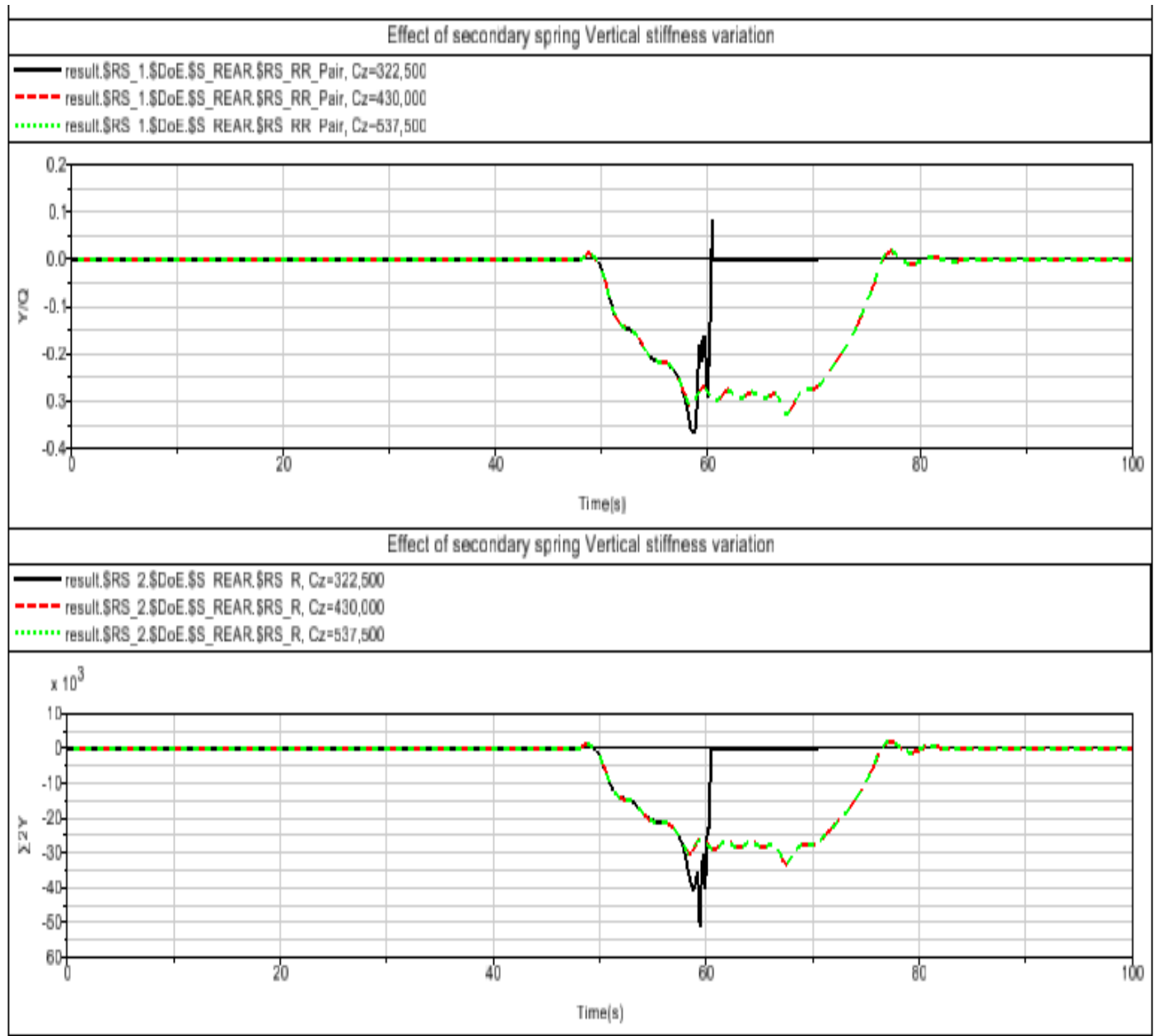


Figure: 4.12: Effect of secondary spring Vertical stiffness variation

In figure: 4.12: due to decreasing up to 15% and increasing up to 25% of original secondary spring Vertical stiffness value both derailment coefficient and ΣY sum rail lateral force have no any change. Therefore, with in this interval secondary spring Vertical stiffness variation have no effect on risk of derailment, but decreasing of secondary spring Vertical stiffness above 15% of original value vehicle derailment occur at curved critical speed running on curved track.

3.3.5 Effect of secondary damper lateral damping variation

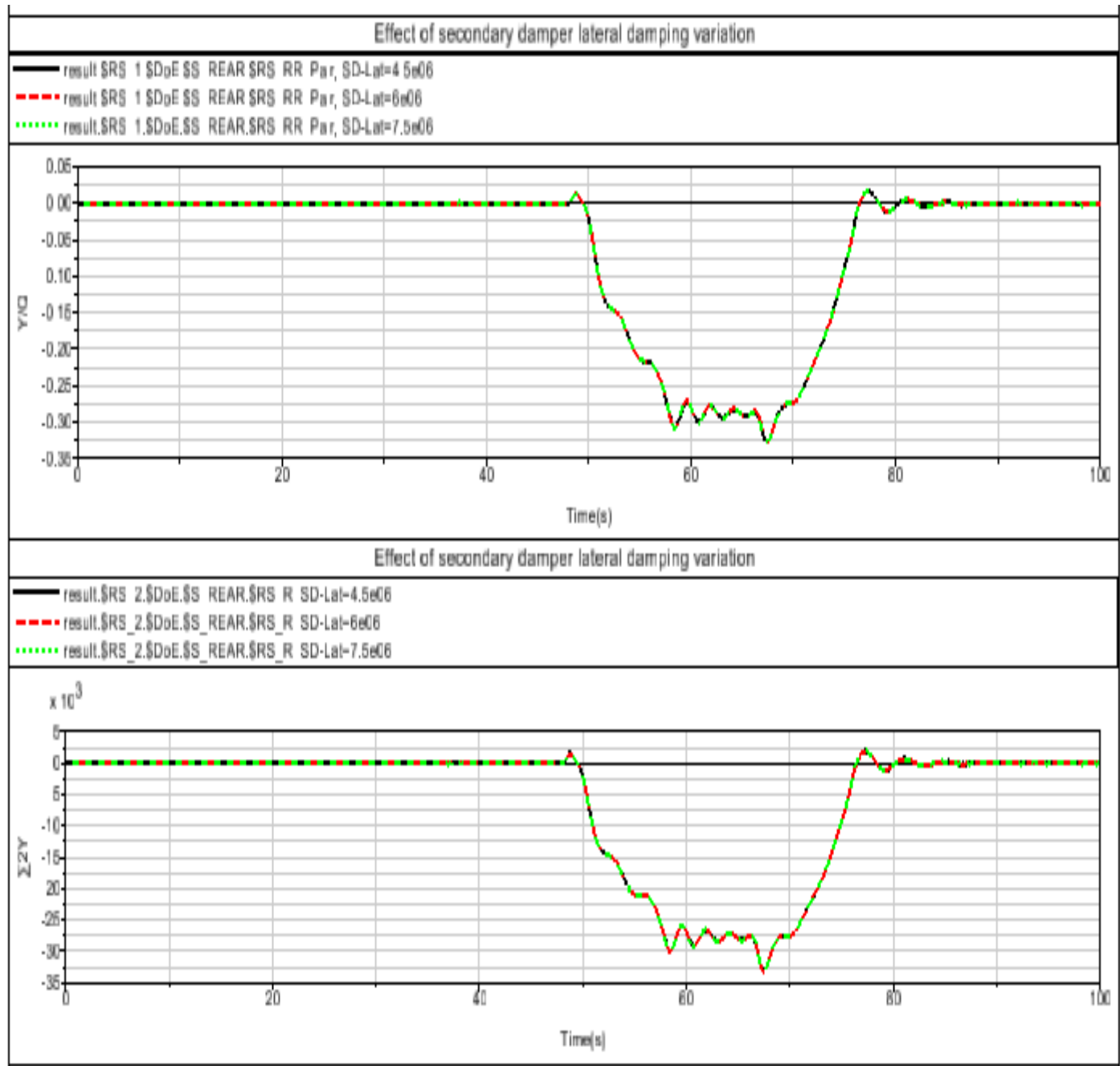


Figure: 4.13: Effect of secondary damper lateral damping variation

In figure:4.13 both derailment coefficient and Y sum rail lateral force have no any change. Therefore, increase or decrease of secondary damper lateral damping variation have no effect on risk of derailment of vehicle running on curved track.

3.3.6 Effect of secondary damper vertical damping variation

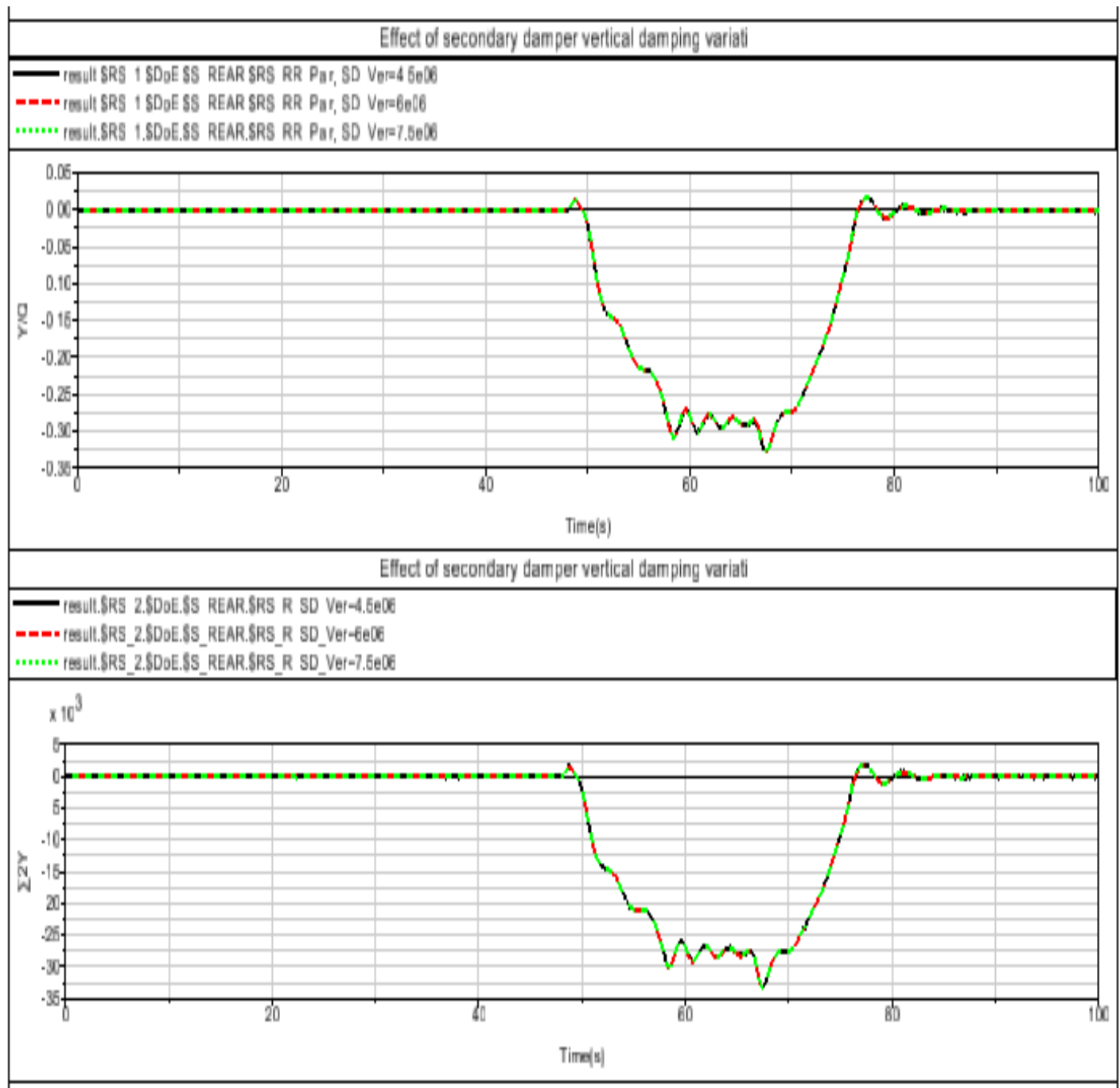


Figure: 4.14: Effect of secondary damper vertical damping variation

In Figure: 4.14: both derailment coefficient and Y sum rail lateral force have no any change. Therefore, increase or decrease of secondary damper vertical damping variation have no effect on risk of derailment of vehicle running on curved track.

3.3.7 Effect of Bogie wheel base (center to center distance of wheel sets in a bogie)

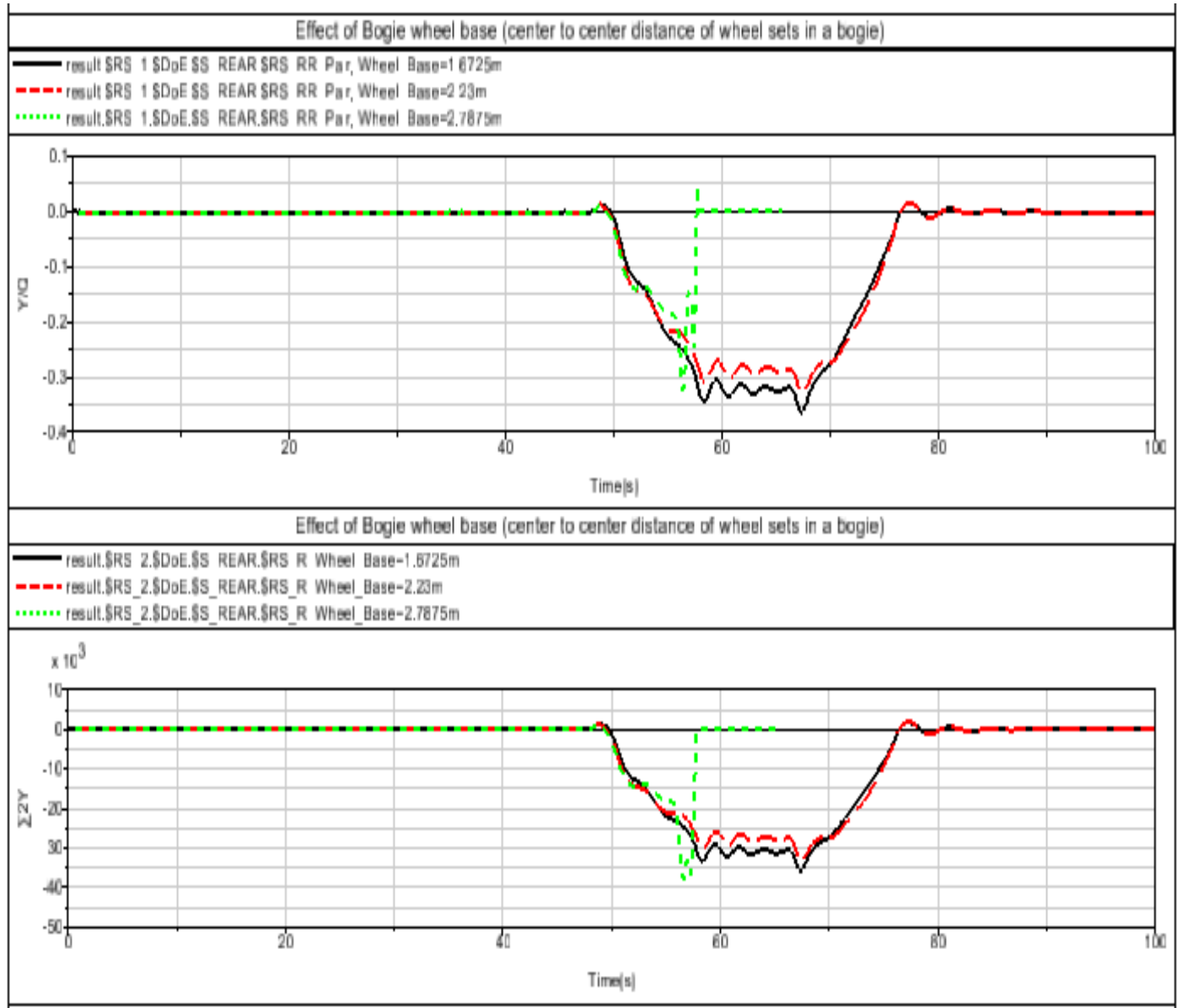


Figure: 4.15: Effect of Bogie wheel base (center to center distance of wheel sets in a bogie)

In figure:4.15 show that both derailment coefficient and Y sum rail lateral force have increase with decreasing wheelbase, but further experiment show in figure :4.16: that more decreasing the value decrease both indicator rather than lead to derailment. And increasing 0.12% wheelbase above the original value vehicle derailment occur. Therefore, decrease of wheelbase safe to

derailment, but increasing of wheelbase lead to vehicle derailment at curved critical speed. Further experiment show that decreasing 25% from original value improve the critical speed by 14.4km/h.

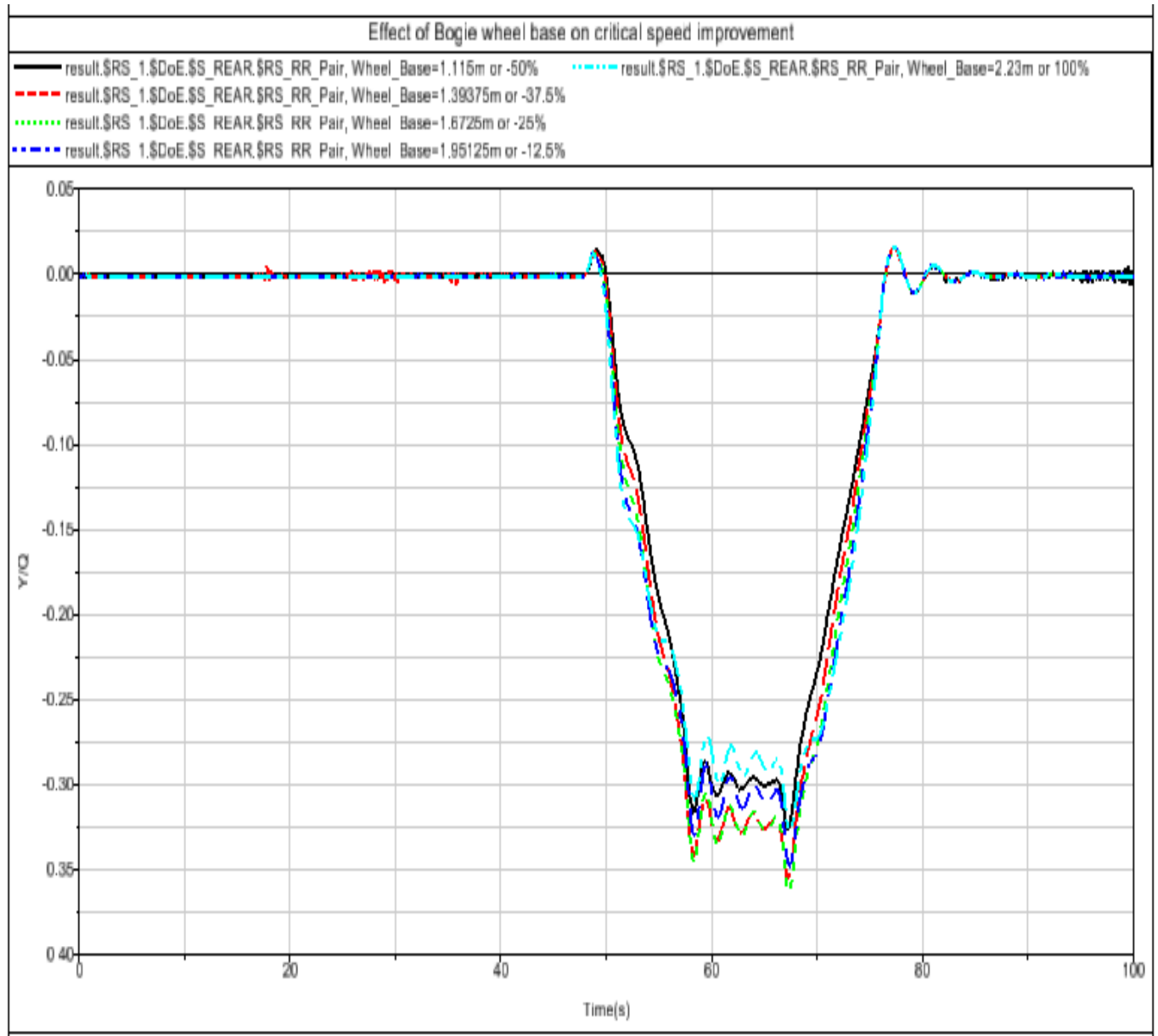


Figure: 4.16: Effect of Bogie wheel base on critical speed improvement

3.3.8 Effect of wheel Radius

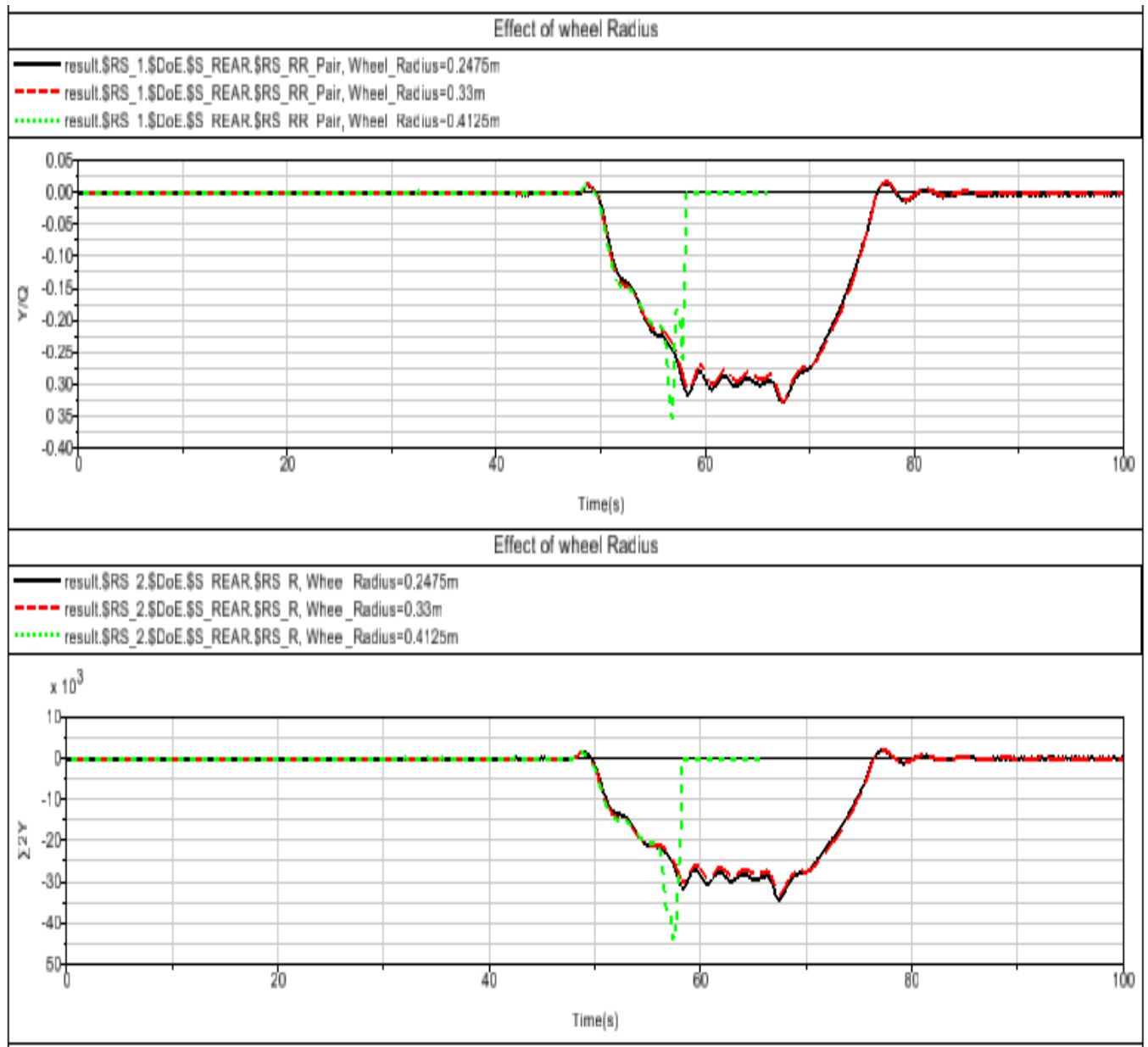


Figure: 4.17: Effect of wheel Radius

In the above figure: 4:17: both derailment coefficient and Y sum rail lateral force have decrease with decreasing wheel radius and increasing 0.12% wheel radius above the original value vehicle derailment occur.. Therefore, decrease of wheel radius safe to risk of derailment, but increasing of wheel radius lead to derailment. Further experiment show that decreasing 25% from original value improve the critical speed by 10km/h.

Chapter Five

5. Conclusion, Recommendation and Future Work

5.1 Conclusion

To study the effect of railway vehicle parameters on derailment, to have reliable engineering computational models. To be achieved this purpose, the MBS software SIMPACK v9.6 railway module is utilized to create a simple but reliable conventional railway vehicle model as well as to perform the analysis of its dynamics behavior. A 41 degrees-of-freedom (DOFs) railway vehicle model created in SIMPACK composed of two bogie frames connected to four wheelsets and a carbody frame by means of the primary and the secondary suspension components, respectively. Parameter study have been performed in order to asses the parameters variation effect on railway vehicle derailment running on curved track at curved track critical speed. The parameters are carbody parameters(mass, rolling mass moment of inertia I_{xx} , pitch mass moment of inertia I_{yy} , yaw mass moment of inertia I_{zz} and vertical mass center), secondary suspension parameters, wheelbase and wheel radius. Each parameters in the study had to varying 25% upper and lower of original designed value. The simulation carried out firstly by determining curved track critical speed and evaluation of each parameter variation effect to derailment at this speed summarized as follow:

- Simulation results demonstrate that increasing carbody mass, yaw mass moment of inertia(I_{zz}) of carbody, vertical mass center of carbody, secondary spring longitudinal shear stiffness, secondary spring lateral shear stiveness, secondary spring vertical shear stiffness, bogie wheelbase and wheel radius above the original value have risk of derailment. As the same time more decreasing carbody mass below the original have risk of derailment.
- Simulation results demonstrate that decreasing yaw mass moment of inertia(I_{zz}) of carbody, vertical mass center of carbody, secondary spring longitudinal shear stiffness, bogie wheelbase, wheel radius below the original value have safe to risk of derailment.
- From SIMPACK further experiment decreasing 25% of orignal value of wheel base, wheel radius, vertical mass center of carbody, secondary spring longitudinal shear stiffness and

yaw mass moment of inertia(I_{zz}) of carbody improve curved track critical speed by 14.4Km/h, 10Km/h, 3.6 Km/h, 1.8Km/h and 1 Km/h respectively.

- Simulation results demonstrate that increasing or decreasing of rolling mass moment of inertia(I_{xx}) of carbody, pitch mass moment of inertia(I_{yy}) of carbody, secondary vertical damper damping and secondary lateral damper damping, decreasing secondary spring lateral shear stiffness, decreasing secondary spring vertical shear stiffness have no effect to risk of derailment.

5.2 Recommendation

The railway vehicle parameters variation research analyzes only the safety running behavior of the railway vehicle running on curved track, this means in the study was not included other running behavior such as ride comfort and wear. These parametric variation may have negative effect on other running behavior. Therefore decreasing yaw mass moment of inertia(I_{zz}) of carbody, vertical mass center of carbody, secondary spring longitudinal shear stiffness, bogie wheelbase, wheel radius below the original value have safe to risk of derailment and improve curved track critical speed, but to take the parameters variation as positive the effect of those variation will investigate to other running behavior, must be improved or within the permissible limit.

5.3 Future Work

From the present study there are some good achievements were obtained, there are other works remaining which to obtain a complete parametric variation analysis, so the work done the following works can be proposed as future work:

- Perform the parametric variation analysis by considering all other running behavior, i.e ride comfort, wear etc.
- The present study simulation done on a model developed by SIMPACK rigid carbody, bogie frame, axle, wheel etc, to get more accurate result the model will first develop/design with finite element software like ansys software or use flexible body and import to SIMPACK.
- Perform the parametric variation analysis including effect of cross wind force on derailment running on curved track.

6. REFERENCE

1. A.H Wickens, Fundamentals of Rail Vehicle Dynamic guidance and stability / A.H. Wickens.p. cm (Advances in engineering; 6), Netherlands, Taylor & Francis e-Library, 2005.
2. Ahmed A Shabana, Khaled E. Zaazaa and Hiroyuki Sugiyama, Railroad Vehicle Dynamics A Computational Approach, United States of America on acid-free paper. Taylor & Francis Group, LLC, 2008.
3. BEATRIZ TENA VILLAR. "Railway Design, Simulation and Analysis by ADAMS/ Rail Software." Master's Thesis June, 2011
4. ADRIÁN HERRERO. "Towards optimization of a high speed train bogie primary suspension." Master's Thesis in the International Master's programmes Applied Mechanics, CHALMERS UNIVERSITY OF TECHNOLOGY. Sweden 2013
5. Christoph Weidemann."State-of-the-Art Railway Vehicle Design with Multi-Body Simulation." Gilching, Germany Journal of Mechanical systems for Transportation and Logistics, Review Paper. 2010,
6. Rail Transport in Ethiopia. Wikipedia, the Free Encyclopedia.
http://en.wikipedia.org/wiki/Rail_transport_in_Ethiopia, 2012.Accessed 23/09/14
7. Iwnicki and A. H Wickens. Validation of a matlab railway vehicle simulation using a scale roller rig. Vehicle System Dynamics, 30, 1998.
8. SIMPACK (9.6): SIMPACK Documentation library, 2014.
9. Saeed Hossein Nia , Thesis on dynamic modelling of freight wagons, Department Mechanical Engineering Belkinge Institute of Technology, Karlskrona, Sweden, 2011
10. D. Ramy Elsayed Shaltout, thesis on Multibody Approach For Railway Dynamic Analysis, Department of Mechanical and Material Engineering UNIVERSITAT POLITÈCNICA DE VALÈNCIA, 2013
11. Escalona, J. and R. Chamorro Efficient On-Line Calculation of the Wheel-Rail Contact Forces in Multibody Dynamics. Symposium of Advances in Contact Mechanics. Delft, Netherlands, 2008.
12. Vivek Kumar , Investigation of vertical dynamic behavior and modelling of a typical Indian rail road vehicle through bond graph, Department of Mechanical Engineering, SLIET Longowal, Sangrur, Punjab, India, 2009.

13. Bingrong Miao, Weihua Zhang, Jihui Zhang, and Dingchang Jin, Evaluation of Railway Vehicle Car Body Fatigue Life and Durability using Multi-disciplinary Analysis Method, State Key Laboratory of Traction Power, Southwest Jiaotong University, Chengdu 610031 Sichuan, P.R. China, 2009
14. D. Brabie & E. Andersson, Dynamic simulation of derailments and its consequences, Division of Railway Technology Royal Institute of Technology (KTH) , Stockholm, Sweden Published online: 04 Apr 2007
15. Oldrich Polach and Andreas Böttcher, A new approach to define criteria for rail vehicle model validation, Bombardier Transportation (Switzerland) AG, Zürcherstrasse 39, CH-8401 Winterthur, Switzerland;
Alstom Transport Deutschland GmbH, Linke-Hofmann-Busch Str. 1, D-38239 Salzgitter, Germany: 4 January 2014.
16. Berta Suarez a , Jesus Felez a , Joaquin Maroto a & Pablo Rodriguez, Sensitivity analysis to assess the influence of the inertial properties of railway vehicle bodies on the vehicle's dynamic behavior, Vehicle System Dynamics: International Journal of Vehicle Mechanics and Mobility, Madrid , Spain Published online: 24 Sep 2012.
17. Mohammad Reza Ghazavi* and Majid Taki, Dynamic simulations of the freight three-piece bogie motion in curve, Department of Mechanical Engineering, Faculty of Engineering, Tarbiat Modares University, Tehran, Iran, 6 October 2007.
18. Huimin Wu and Nicholas Wilson, Handbok of railway Vehicle dynamic
19. Joanna Moody, Critical Speed Analysis of Railcars and Wheelsets on Curved and Straight Track, Presented to the Department of Physics and Astronomy Bates College, Lewiston, Maine April 1, 2014.
20. Hugo Miguel Pacheco Magalhães, Development of Advanced Computational Models of Railway Vehicles, TECNICO LISBOA, 2013
21. Nicholas Wilsona*, Robert Friesa, Matthew Wittea, Andreas Haigermoserb, Mikael Wrangc, Jerry Evansd and Anna Orlovae, Assessment of safety against derailment using simulations and vehicle acceptance tests: a worldwide comparison of state-of-the-art assessment methods, Taylor & Francis Informa Ltd Registered in England and Wales, 24 Jun 2011

Appendices

Appendix: A: Vehicle Modelling Data

Most design data of the rail vehicle are collected from SIMPACK model data base from Internet, i.e. WWW.simpack.com/modeldatabase/rail

Table: Appendix: A.1: Modeling data of rigid body and wheel-rail

Subvar	Railway vehicle Parameter	Value
\$_Wheelset_Mass	Mass of Wheelset	1831Kg
\$_Wheelset_Ixx	Roll mass moment of inertia of Wheelset	1120 Kg.M ²
\$_Wheelset_Iyy	Pitch mass moment of inertia of Wheelset	112 Kg.M2
\$_Wheelset_Izz	Yaw mass moment of inertia of Wheelset	1120 Kg.M2
\$_Wheelset_CM_Z	Vertical position of wheelset Center of Gravity	0.0m
\$_Bogie_Mass	Mass of Bogie	2615Kg
\$_Bogie_Ixx	Roll mass moment of inertia of Bogie	1722Kg.M2
\$_Bogie_Iyy	Pitch mass moment of inertia of Bogie	1476Kg.M2
\$_Bogie_Izz	Yaw mass moment of inertia of Bogie	3076Kg.M2
\$_Bogie_CM_Z	Vertical position of Bogie Center of Gravity	-0.47m
\$_Bogie_CM_Z	Vertical position of bogie Center of Gravity	-0.60m
\$_Carbody_Mass	Mass of Carbody	32,000Kg
\$_Carbody_Ixx	Roll mass moment of inertia of Carbody	56800Kg.M2
\$_Carbody_Iyy	Pitch mass moment of inertia of Carbody	19700000Kg.M2
\$_Carbody_Izz	Yaw mass moment of inertia of Carbody	19700000Kg.M2
\$_Carbody_CM_Z	Vertical position of Carbody Center of Gravity	-1.67m
\$_Bogie_Base	Bogie Center to Center Distance	19m
\$_Wheel_Radius	Wheel Nominal Radius	0.33m
\$_Rail_Cant	Rail cant	1:40
Rail Guage	Rail Guage	1.435m
\$_Friction	Wheel/rail coefficient of friction	0.4
\$_Wheel_base	Wheel base	2.23

Table: Appendix: A: 2: Modeling data of Suspension

Component	Nom- Length (m)	Type	Stiffness			Damping		
			cx	cy	cz	dx	dy	dz
Primary Spring	0.42	Transl. Ser	1.5x10 ⁴	2x10 ³	0.00	6x10 ⁷	7.5x10 ⁶	0.00
		Transl. Par	0.00	0.00	0.00	3.14x10 ⁷	3.88x10 ⁶	1.22x10 ⁶
Primary Damper	0.42	Vertical	0.00	0.00	6x10 ⁵	Figure Appendix:A.2:		
Secondary Spring	0.475	Shear	8x10 ⁴	8x10 ⁴	2.15x10 ⁵	0.00	0.00	0.00
		Roll bending	105000Nm/rad					
		Pitch bending	105000Nm/rad					
Secondary Damper	0.475	Vertical	0.00	0.00	6x10 ⁶	Figure Appendix:A.2: and A:3		
		Lateral	0.00	0.00	6x10 ⁶			
Bump Stop	Spring –Damper serial		Figure Appendix:A.3:					
Anti roll bar	Rotational Spring-damper		About x axis = 940,000Nm/rad					
Traction Rod	1.12	Spring	5x10 ⁶	0.00	0.00	0.00	0.00	0.00
	1.12	Damper	10x10 ⁶	0.00	0.00	2.5x10 ⁴	0.00	0.00

Table: Appendix: A.3: Wheel and rail material properties and contact data

Type of Propertie or Data	Value
Material Young's Modules	2.1×10^{11}
Poisson number	0.28
Contact reference damping	10×10^4
Equivalent conicity lambda	0.186
Contact angle coefficient epsilon	15:81
Roll angle coefficient sigma	0.0372

Figure: Appendix: A: 1: Assigned SubVar

\$Bolster_BS_Z	\$PS_cs_x	\$PS_cs_x
\$Bolster_ARB_Z	\$PS_cs_y	\$PS_cs_y
\$Bolster_CAR_Z	\$PS_cpx	\$PS_cpx
\$Sec_AntiRoll_c	\$PS_cpy	\$PS_cpy
\$Tra_Rod_Spring_C	\$PS_cpz	\$PS_cpz
\$Tra_Rod_Damp_d	\$PD_Vert_c	\$PD_Vert_c
\$Tra_Rod_Damp_c	\$SS_cx	\$SS_cx
\$SD_Vert_c	\$SS_cy	\$SS_cy
\$SD_Lat_c	\$SS_cz	\$SS_cz
\$Bogie_Base	\$SS_cr	\$SS_cr
\$Wheel_Base	\$SS_cp	\$SS_cp
\$Bogie_Fx	\$PS_NOM_Z	\$PS_NOM_Z
\$Bogie_Rx	\$Wheelset_Mass	\$Wheelset_Mass
\$Bogie_mass	\$Wheelset_Ixx	\$Wheelset_Ixx
\$Bogie_Ixx	\$Wheelset_Iyy	\$Wheelset_Iyy
\$Bogie_Iyy	\$Wheelset_Izz	\$Wheelset_Izz
\$Bogie_Izz	\$SS_F	\$SS_F
\$Bogie_CM_X	\$Bogie_SS_Fx	\$Bogie_SS_Fx
\$Bogie_CM_Y	\$Bogie_SS_Rx	\$Bogie_SS_Rx
\$Bogie_CM_Z	\$Wheel_Radius	\$Wheel_Radius
\$Wheelset_SS_Ry	\$Friction_C	\$Friction_C
\$Wheelset_PS_Ry	\$Bogie_L1	\$Bogie_L1
\$Wheelset_PS_Ly	\$Bogie_L2	\$Bogie_L2
\$Bolster_Mass	\$Bogie_H1	\$Bogie_H1
\$Bolster_Ixx	\$Bogie_H2	\$Bogie_H2
\$Bolster_Iyy	\$Bogie_B1	\$Bogie_B1
\$Bolster_Izz	\$Bogie_Frame_Width	\$Bogie_Frame_Width
\$Axle_Z	\$Axle_Diameter	\$Axle_Diameter
\$PS_ds_x	\$Axle_Length	\$Axle_Length
\$PS_ds_y	\$Bogie_Z	\$Bogie_Z
\$PS cs x	\$Bolster Top Z	\$Bolster Top Z

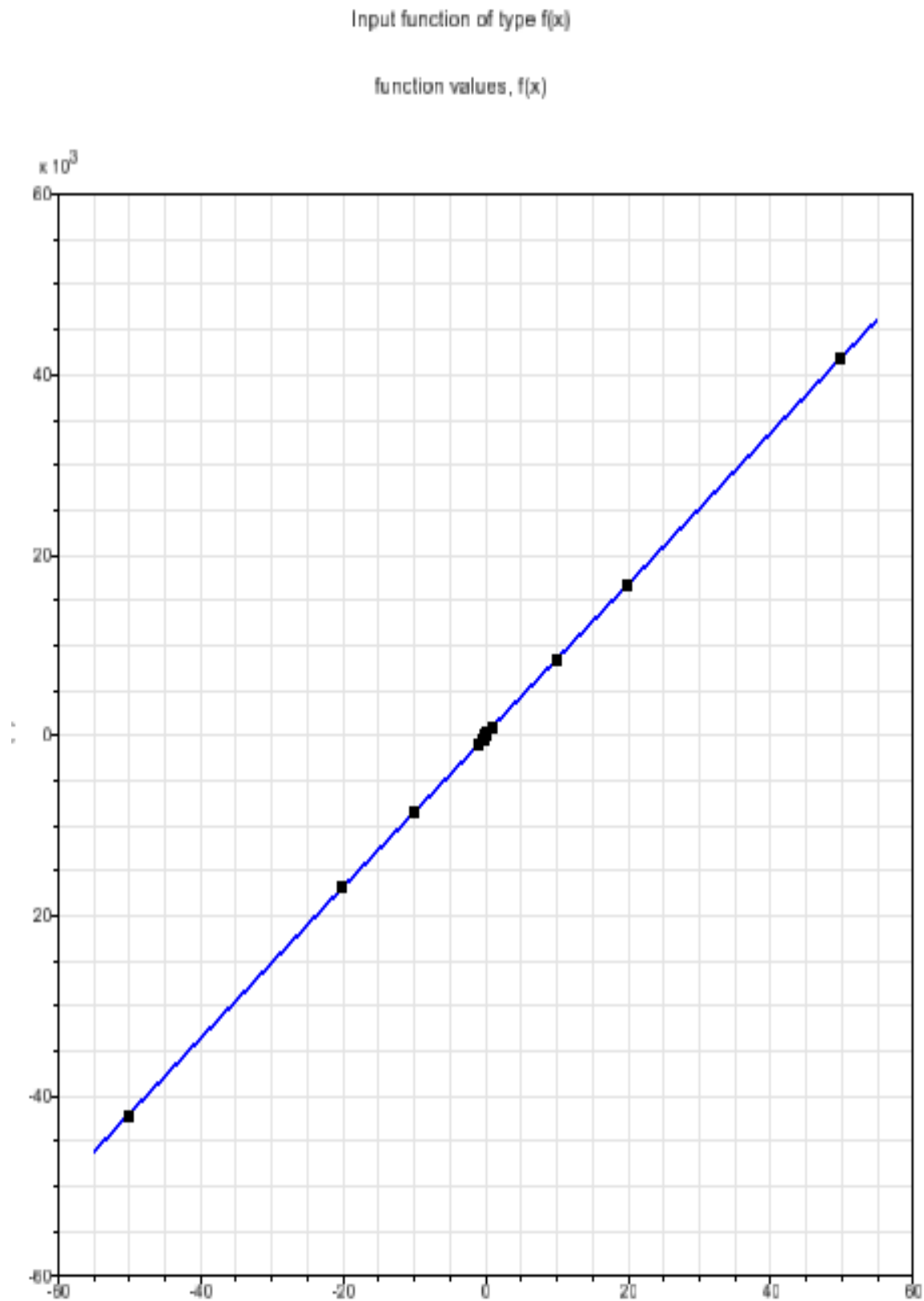
Figure Appendix: A.2: Characteristics of primary damper vert. damping

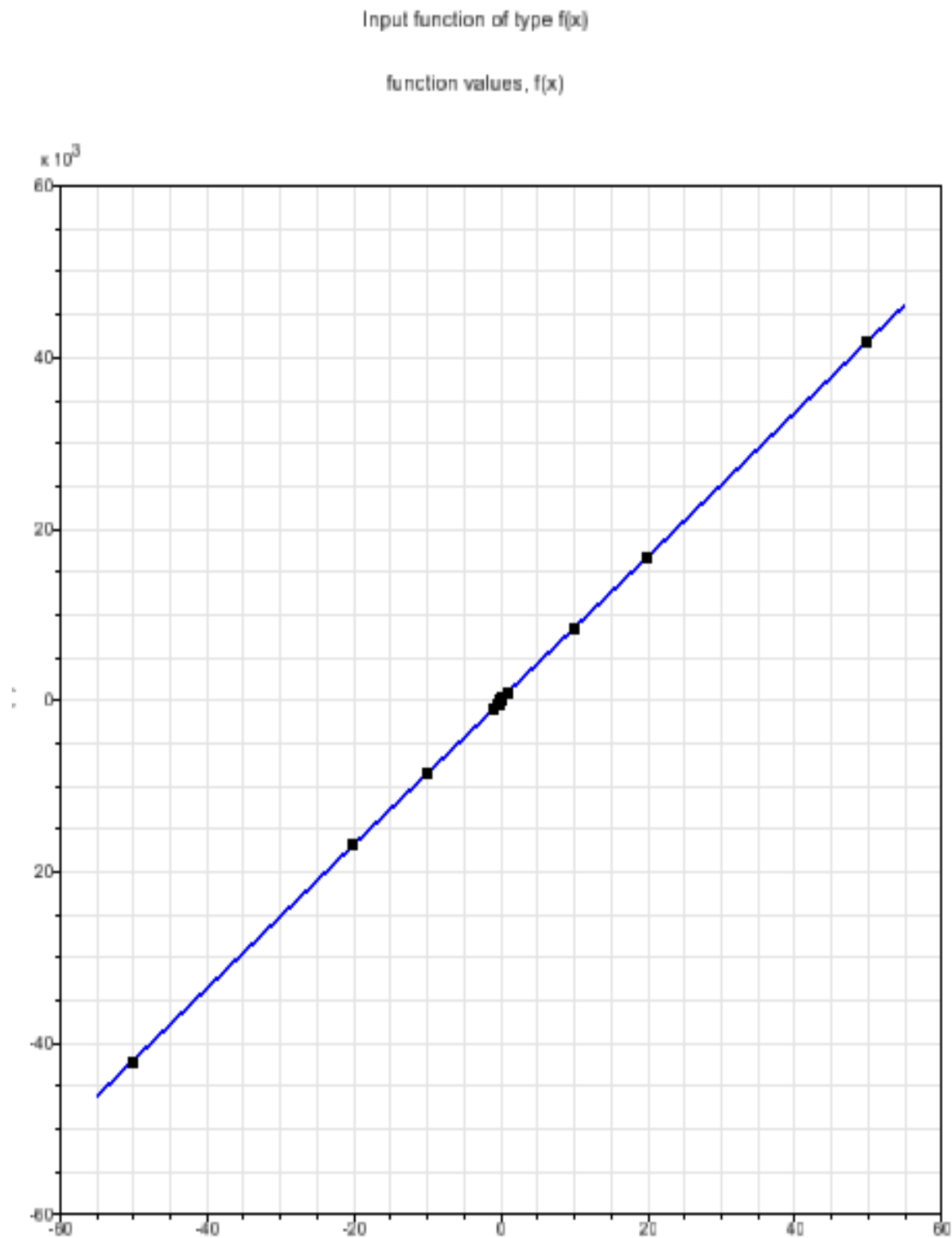
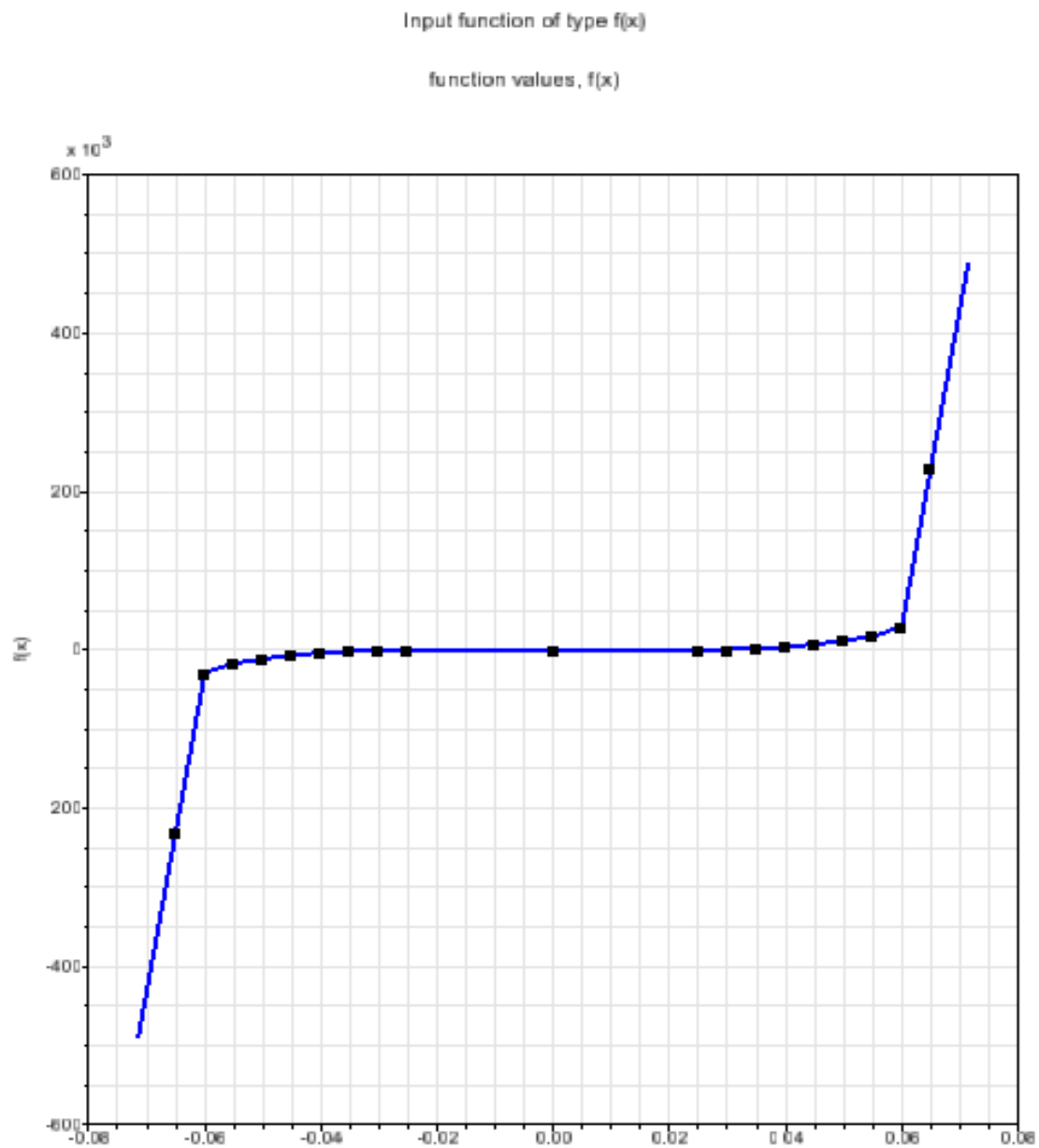
Figure Appendix: A.3: Characteristics of secondary damper vert. damping

Figure Appendix: A.4: Characteristics of bump stop

Appendix: B: Other Figures

Figure: Appendix: B:1: Preload Calculation Dialog Box

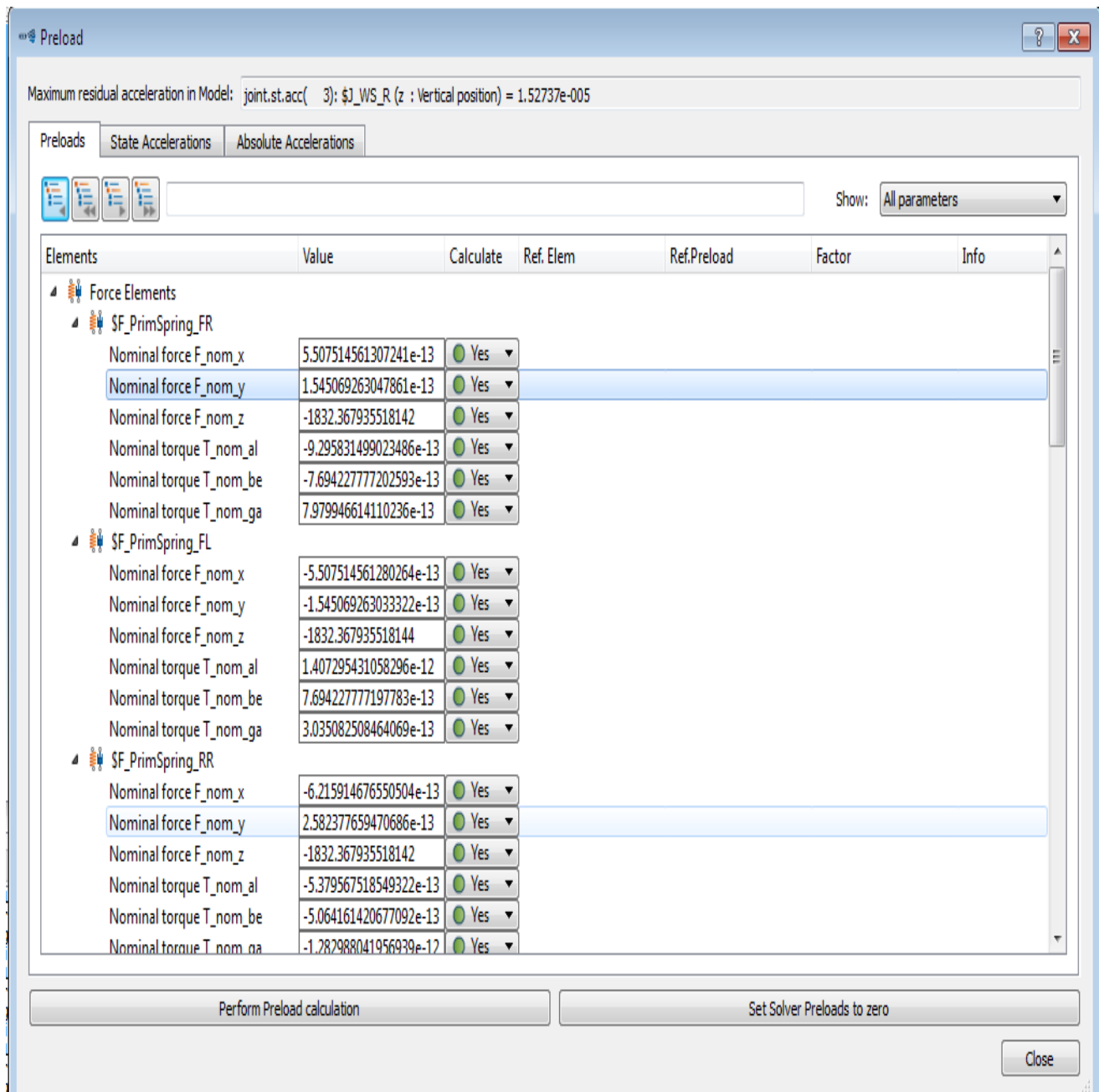


Figure: Appendix: B.2: Static Equilibrium Dialog Box